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JUL 13 1944

MACHINE DESIGN

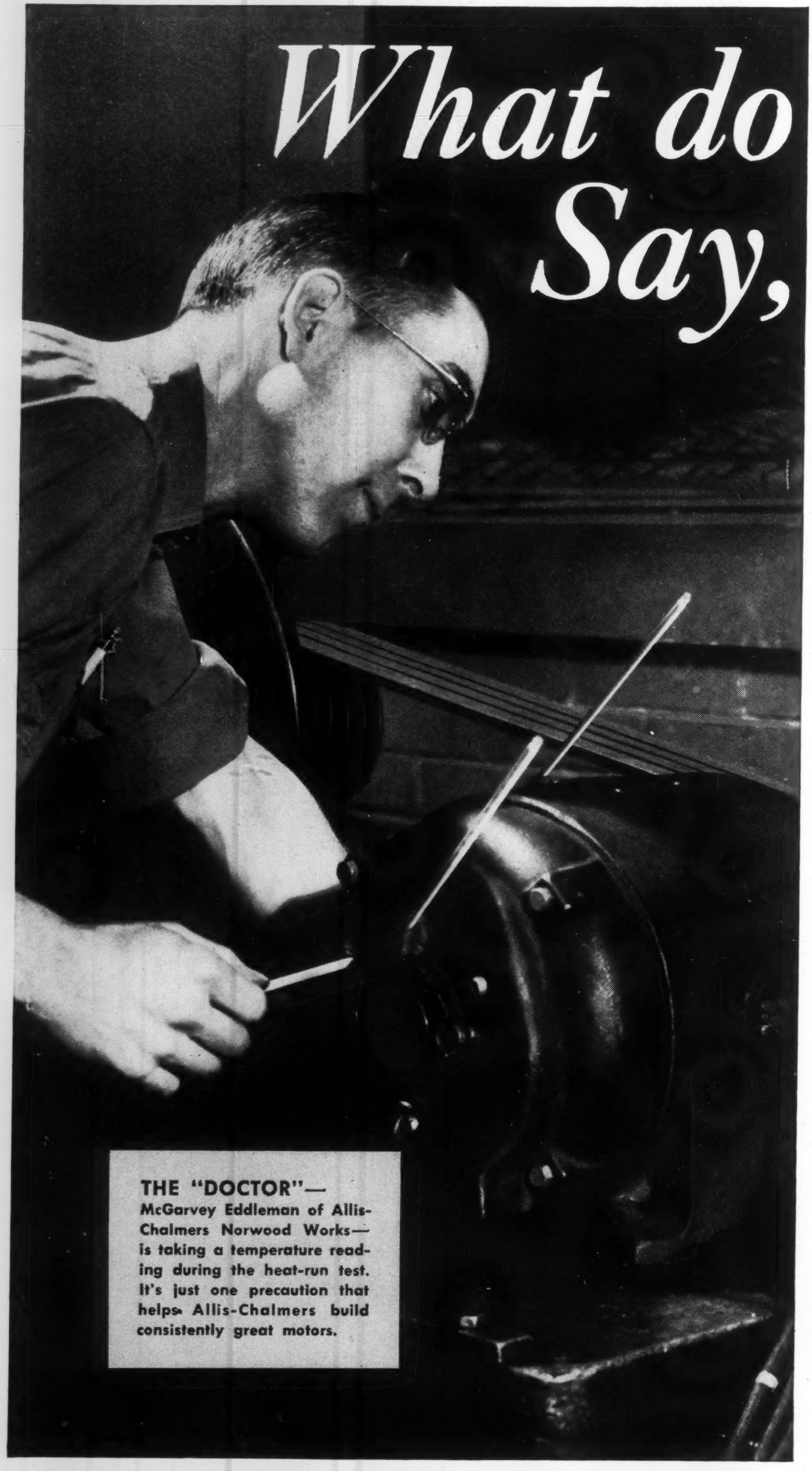
July 1944

In This Issue:

Jet Propulsion

Hydraulic Analogies

What do You Say, Doc?



THE "DOCTOR"—
McGarvey Eddleman of Allis-Chalmers Norwood Works—is taking a temperature reading during the heat-run test. It's just one precaution that helps Allis-Chalmers build consistently great motors.

NO—THAT'S NOT a sick motor. Actually it's a very *healthy* one. But that's something we must know — through generous sample testing — before Allis-Chalmers motors can tackle jobs for you.

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• • •

If you could meet and talk with the men who build Allis-Chalmers motors, you might be surprised to learn how keenly they are aware of the big *personal* stake they have in every motor they build for you.

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YOU CAN DEPEND ON ALLIS-CHALMERS MOTORS

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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MAIN OFFICE: The Penton Publishing Co., Penton Building, Cleveland 13.

BRANCH OFFICES: *New York* 17, 16 East 43rd St., *Chicago* 11, 520 N. Michigan Ave.; *Pittsburgh* 19, Koppers Building; *Detroit* 2, 6560 Cass Ave.; *Washington* 4, National Press Building; *Los Angeles* 4, 130 North New Hampshire Ave.; *London S.W.* 1, 2 Caxton St., Westminster.

PUBLISHED BY The Penton Publishing Co. E. L. Shaner, Pres. and Treas.; G. O. Hays, Vice Pres. and Gen. Mgr.; R. C. Jaenke, Vice Pres.; F. G. Steinebach, Vice Pres. and Secy.; E. L. Werner, Asst. Treas. Published on seventh of month. Subscription in U.S. and possessions, Canada, Cuba, Mexico, Central and South America: Two years, $10; one year, $6. Single copies, 50 cents. Other countries: Two years, $14; one year, $8.

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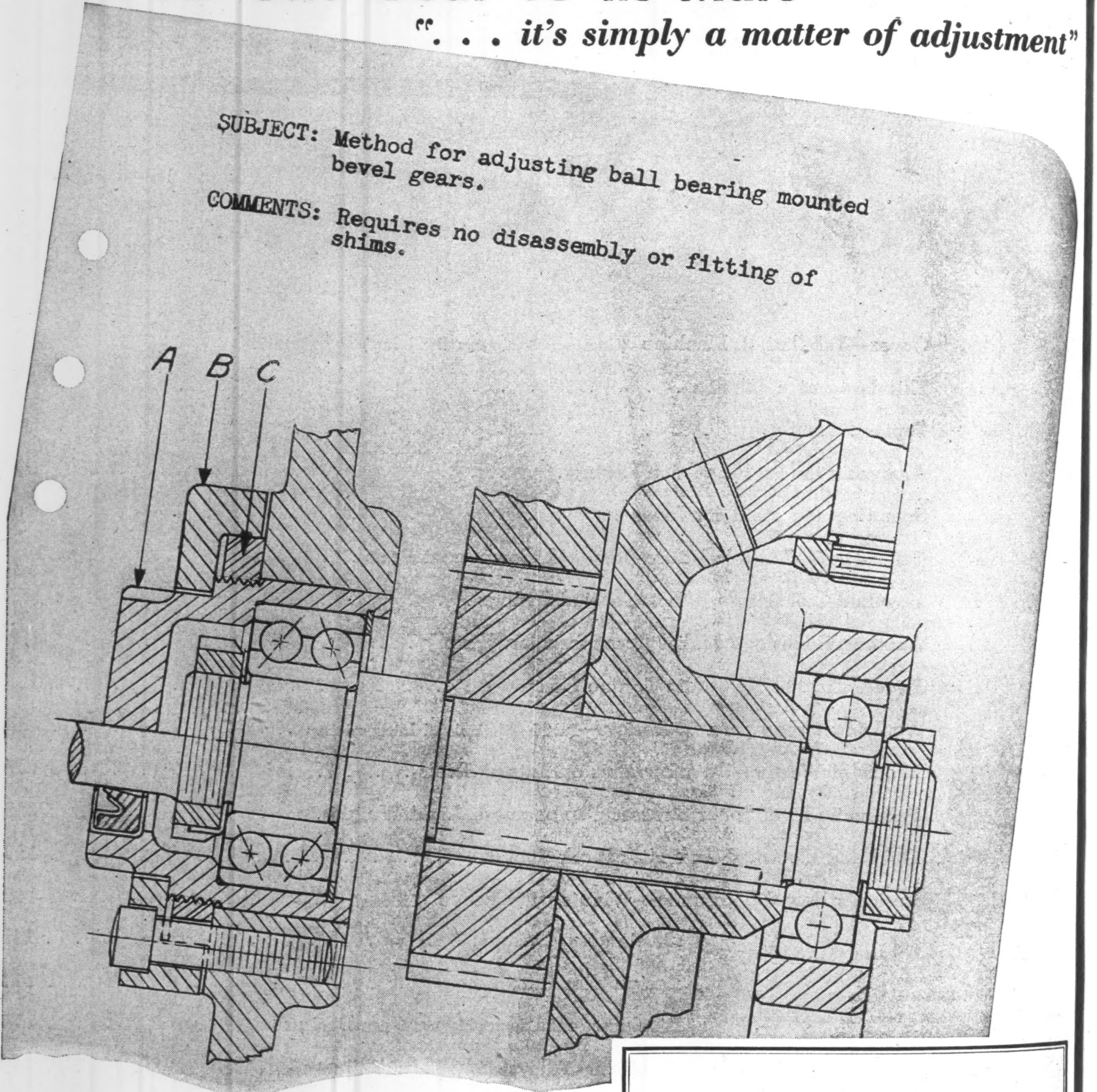
Acceptance under act of June 5, 1934, authorized July 20, 1943.

As One Gear To Its Mate—

". . . it's simply a matter of adjustment"



SUBJECT: Method for adjusting ball bearing mounted bevel gears.

COMMENTS: Requires no disassembly or fitting of shims.

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Gives stepless gear adjustment to any degree of exactness required.

No removal of parts necessary. Simply loosen bolts slightly so sleeve "A" can be turned amount desired. Tighten bolts to clamp cover "B." Nut "C" is notched for bolts; does not rotate.

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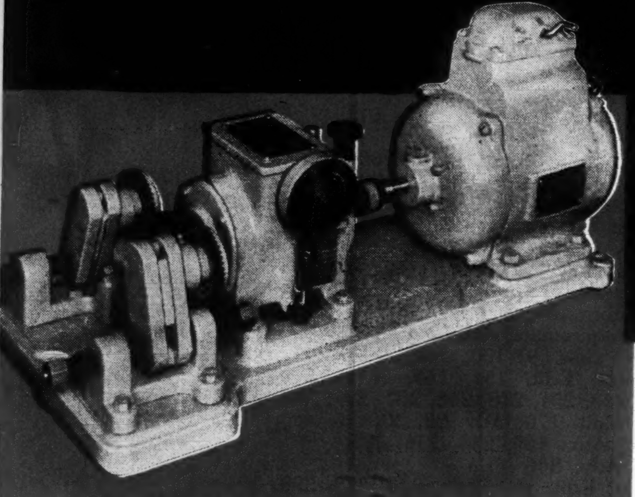
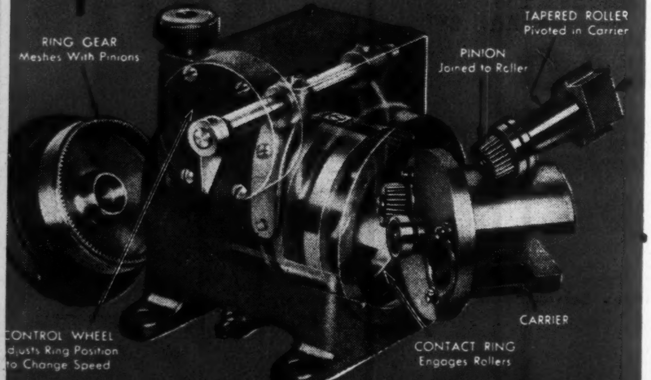
For POSITIVE-TYPE PUMPS AS IN THE ZENITH



EXACT METERING and PROPORTIONING

are the requirements met by the Zenith Metering Pump. It was primarily designed to force the "dope" or spinning solution through the minute hole of a spinnerette in the production of synthetic yarns. The volumetric efficiency of the pump combined with its ability to build up high discharge pressures with surprisingly slight effort has made it adaptable for uses as a part of a viscosimeter, for force feed oilers against high pressures and for constant proportioning. The Graham gives the Zenith infinite variable speed from zero to 220 RPM. It permits sharp speed holding — so vitally important in metering or proportioning.

SEVEN YEARS of performance in important industrial and war applications have proven the soundness of design, the ruggedness and fool-proof operation of the Graham Variable Speed Transmission.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Pivoted in Carrier
PINION
Joined to Roller
CONTROL WHEEL
Adjusts Ring Position to Change Speed
CONTACT RING
Engages Rollers
CARRIER

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has these* IMPORTANT ADVANTAGES

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*2. SHARP SPEED HOLDING

— a vital necessity for the precise metering and proportioning — accomplished by the Zenith.

*3. ALL METAL CONSTRUCTION

Compactness — simplified lubrication — no belts, no tubes. These same features have led to the adoption of the Graham for a great variety of other machines requiring a superior wide-range drive at moderate cost.

Write for Bulletin No. 506.

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This issue at a glance

Jet Propulsion Engines Are Under Wraps . . . .

. . . . but their operating principles aren't. What are these aerodynamic and thermodynamic principles and what computations are involved? How dependent is J.P. on the efficiency of its compressor? "Analyzing the Jet Propulsion Engine", Page 97, will bring your engineering knowledge of this significant development up to date.

How About Turning the Tables . . . .

. . . . on hydraulic analogies of electrical circuits? Why not *electrical* analogies of *hydraulic* or *pneumatic* systems? Perhaps such "cross-analogies" can point the way to new design conceptions. Turn to Page 137 for Elberty's slant on this.

Tailor-Made Electrical Control Assemblies . . . .

. . . . have much to commend them. They're easy to alter or add to and their component parts are readily procured at low cost. Standardized electrical parts facilitate the construction of these tailor-made units greatly. See Page 103.

Current Practice in Spring Design . . . .

. . . . is to use lower working stresses for tension than for compression springs. Why is this justified? And what are the important factors which influence the stressing of spring end coils? If you're designing springs don't miss this article. It's based on a chapter from Dr. Wahl's forthcoming book *Mechanical Springs*. Page 107.

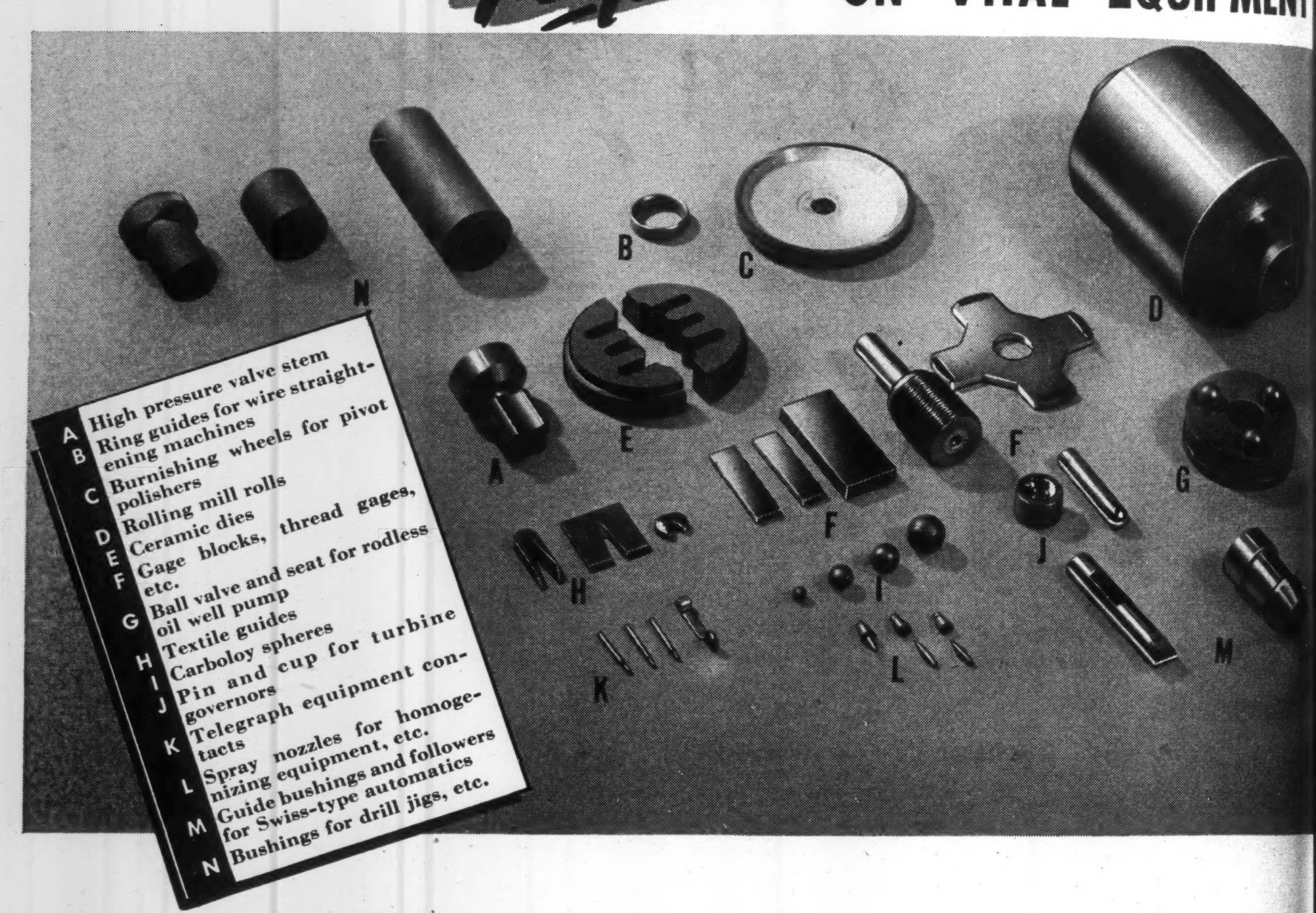
Compactness and Light Weight Aren't the Whole Story . . . .

. . . . in the design of a portable machine. The designer must keep an eye cocked toward the assembly line. In other words the machine must assemble quickly and at low cost. Designing it in units helps. See Page 133.

Creep Can Have Profound Influences . . . .

. . . . on the functioning and life of many machine parts. This is particularly true when they operate at high temperatures. Trend toward more precise requirements in machine components makes the study of creep imperative. Be sure to read Marin's informative article—the first of a series—starting on Page 123.

HOW THE HARDEST METAL MADE BY MAN *Fights Wear* ON VITAL EQUIPMENT



- A High pressure valve stem
- B Ring guides for wire straightening machines
- C Burnishing wheels for pivot polishers
- D Rolling mill rolls
- E Ceramic dies
- F Gage blocks, thread gages, etc.
- G Ball valve and seat for rodless oil well pump
- H Textile guides
- I Carboloy spheres
- J Pin and cup for turbine governors
- K Telegraph equipment contacts
- L Spray nozzles for homogenizing equipment, etc.
- M Guide bushings and followers for Swiss-type automatics
- N Bushings for drill jigs, etc.

In many diversified fields throughout industry you'll find Carboloy Cemented Carbide being employed as a non-cutting, wear-resistant material to increase the life of equipment subject to excessive wear. A few typical applications are shown above.

Cemented carbide, the hardest metal made by man, has extreme resistance to abrasion, and many forms of corrosion. It retains original size and shape and continues to function efficiently for periods often as long as 100 times that of other metals. It drastically reduces machine downtime, eliminates frequent part replacement, and decreases rejects or scrap.

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For the equipment you *use*, and the products you *sell*—step up performance by placing Carboloy Cemented Carbide at areas of excessive wear.

TO STOP EXCESSIVE WEAR
on vital parts of your product

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TOUGH MOTOR JOB? . . . AND HOW!

"Double the horsepower, increase pumping capacity 50%, cut the weight to one-third."

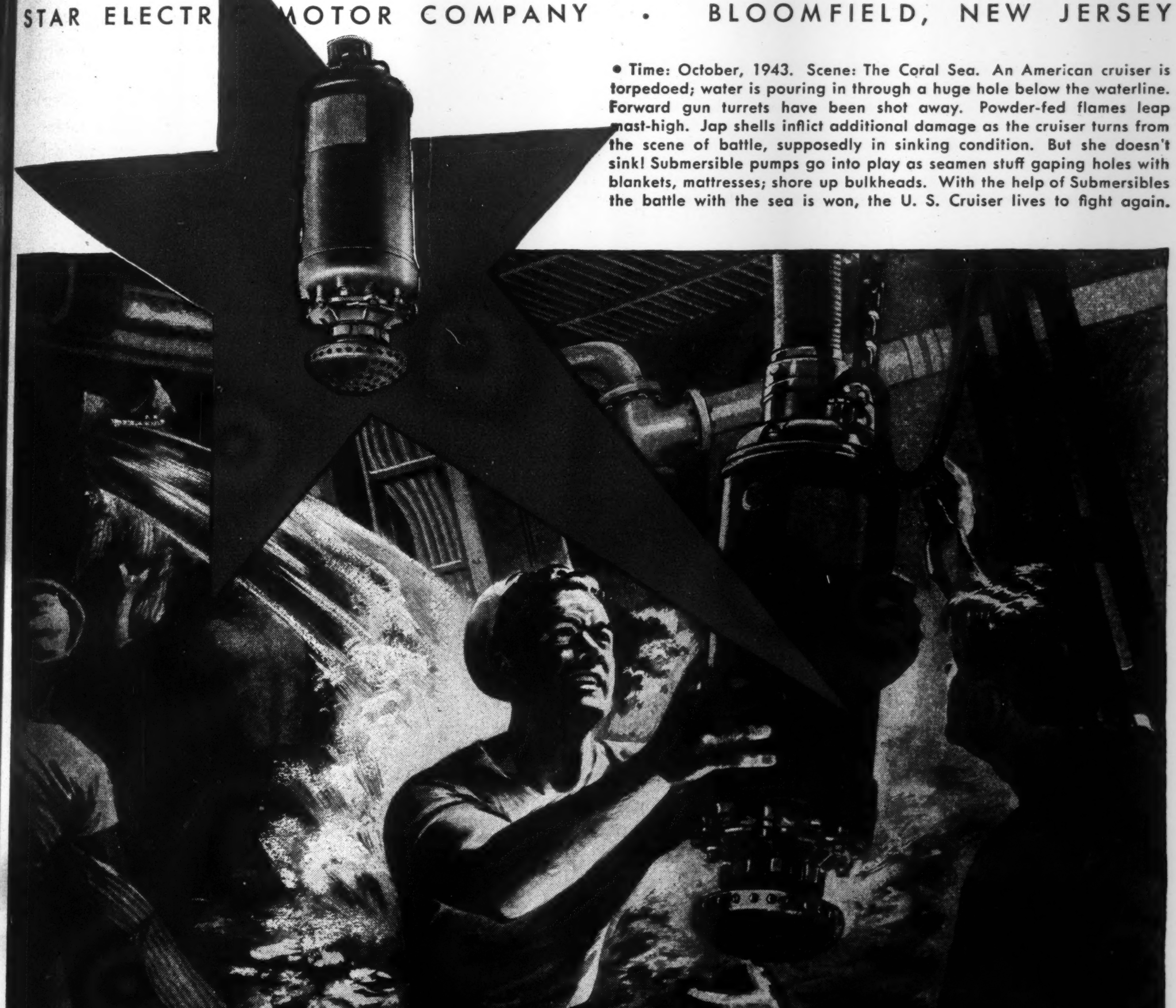
These were the requirements when Star resumed production of submersible pumps for the Navy at the outbreak of the war. Developed by Star after the first World War, pumps were required to deliver water at a rate of 135 gallons per minute at a head of 35 feet—to operate only a few hours at a time. And now—*double* the head to 50 feet—*increase* delivery to 180 gallons per minute, provide continuous operation!

Star engineers—Specialists in "TIGHT SPOTS"—met the requirements and surpassed them—with motors that exceed the pumping requirement of 180 gallons per minute—that have operated continuously on test 6271 hours at a time, to pump 75,252,000 gallons of water. That's 314,365 *tons* of water—and still going strong!

Perhaps you'll never need a motor to meet such extreme needs. But isn't it a satisfaction to know that your motor requirements can receive the same STAR engineering?

STAR ELECTRIC MOTOR COMPANY • BLOOMFIELD, NEW JERSEY

• Time: October, 1943. Scene: The Coral Sea. An American cruiser is torpedoed; water is pouring in through a huge hole below the waterline. Forward gun turrets have been shot away. Powder-fed flames leap mast-high. Jap shells inflict additional damage as the cruiser turns from the scene of battle, supposedly in sinking condition. But she doesn't sink! Submersible pumps go into play as seamen stuff gaping holes with blankets, mattresses; shore up bulkheads. With the help of Submersibles the battle with the sea is won, the U. S. Cruiser lives to fight again.



POWER PACKAGED AS YOU NEED IT

STAR MOTORS

Topics

STEEL CABLE is an ideal type of aircraft device: It is a multiple structure, has internal damping and, long before it fails completely, gives an easily detectable indication of potential danger. Furthermore, cables seldom jam, permitting operation by an auxiliary device.

USE OF ELECTRONICS in vulcanizing tires, a wartime development, is enabling the Army to make major tire repairs in combat zones within minutes.

INCREASED production of 70 to 100 times has been obtained with Carboloy cemented carbide dies for punching motor laminations and drawing cylinder sleeves.

MOISTURE-PROOF plastic known as Fosterite will greatly lengthen the life of radio and other communications equipment used by Allied armies. As an impregnant, being almost as fluid as water, it fills completely every tiny space. As a coating it can be applied to apparatus without leaving air gaps through which moisture could seep. This happens because the material, unlike varnish, requires no liquid solvent which would evaporate during the curing process and cause tiny cracks to appear.

MORE THAN HALF the country's population is living in communities classified by WMC as areas of labor surplus or where the industrial employment problem is slight. As of June 1, 42 per cent was in the labor shortage area—20.4 per cent in Group I and 21.6 in Group II.

WEIGHT REDUCTION amounting to as much as 50 per cent is possible in electrical equipment where design limitations are based on insulating temperature. This has been made possible by Westinghouse's development of Silicone varnish which bridges the gap between organic and ceramic type insulating materials. Physically it is similar to conventional resins and varnishes but differs in that it does not decompose readily at 390 degrees Fahr.

ENEMY WEAPONS of practically every known type are now in our possession. More than 4000 tons of these weapons and ammunition have been classified and studied. Firing tests, for example, of German artillery are being made in comparison with our own.

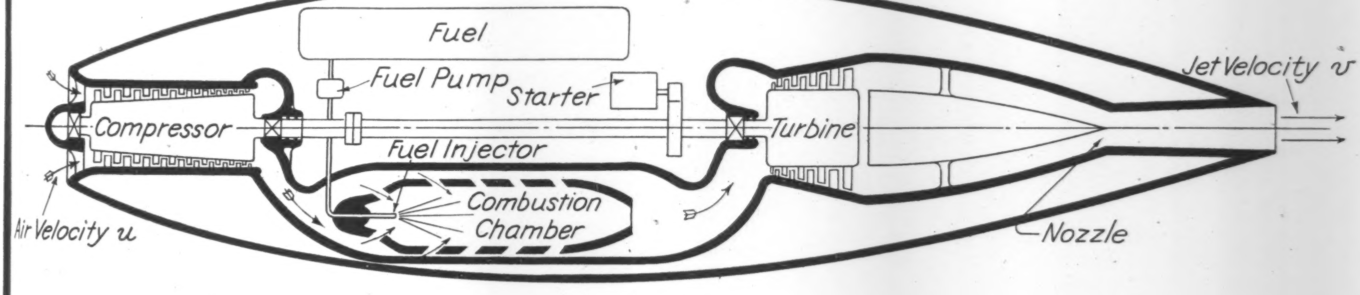
STANDARDIZED grease fittings, developed by the Ordnance Department, have obviated procurement of special types of lubricating guns and facilitated the problem of lubrication under field conditions. The standardization represents an annual saving of approximately $1,500,000.

INVENTORIES of scrap and pig iron are again beginning to decline, carbon scrap showing a larger drop in proportion to the total decrease. Having reached a peak in January 1943 as a result of the scrap drives, the inventories are again approaching the dangerous low of 4,000,000 tons.

GOOD LAYOUT of auxiliary equipment in an airplane will save more weight than the use of the latest alloys and the best possible structure, according to the Douglas Aircraft Co.

CONTINUING NEED for engineering draftsmen to work on construction of maps, strategic posters and graphs and technical plans has been announced by the Civil Service Commission. Women are particularly urged to apply. Application forms may be obtained from first or second-class post offices, the Commission's regional offices or direct from the central office of the Commission, Washington 25, D. C.

MACHINE DESIGN


Fuel
Fuel Pump
Starter
Jet Velocity v
Compressor
Turbine
Fuel Injector
Combustion Chamber
Air Velocity u
Nozzle

ANALYZING the Jet Propulsion Engine

By Colin Carmichael

Fig. 1—Jet propulsion engine, which may be accommodated in a nacelle, consists primarily of a compressor, combustion chamber, gas turbine for driving the compressor, and a nozzle for producing the high-speed jet. Auxiliary fuel injectors may be employed between the turbine and the jet nozzle

SINCE the announcement some months ago that a twin-engined fighter plane employing jet propulsion had been successfully developed, speculation has been rife concerning the details of this radical new powerplant. With the history of the development and the broad features of the jet propulsion principle the general public has become fairly well acquainted through articles in popular magazines. Although no information has been released concerning engineering details, the fundamental principles of J.P. have long been known and may be found in current published material. In an attempt to tie in the new development with these established principles, the following purely speculative discussion is offered. It is hoped that these conjectures will be of interest to engineers and provide a basis for evaluating the possibilities and limitations of jet propulsion.

Development of J.P. has been stimulated by the increasing speed of fighter planes. When air velocity at any point on the surface of a plane or its propeller approaches the speed of sound in air (760 miles per hour at sea level), effect of air compressibility becomes acute, resulting in greatly increased drag. While present planes with level flight speeds in excess of 400 miles per hour still are considerably below the speed of sound, air speeds encountered at certain points in the flow over airfoil sections to gain the necessary lift on the wings and thrust on the propellers may closely approach the speed of sound even when plane speeds are considerably less.

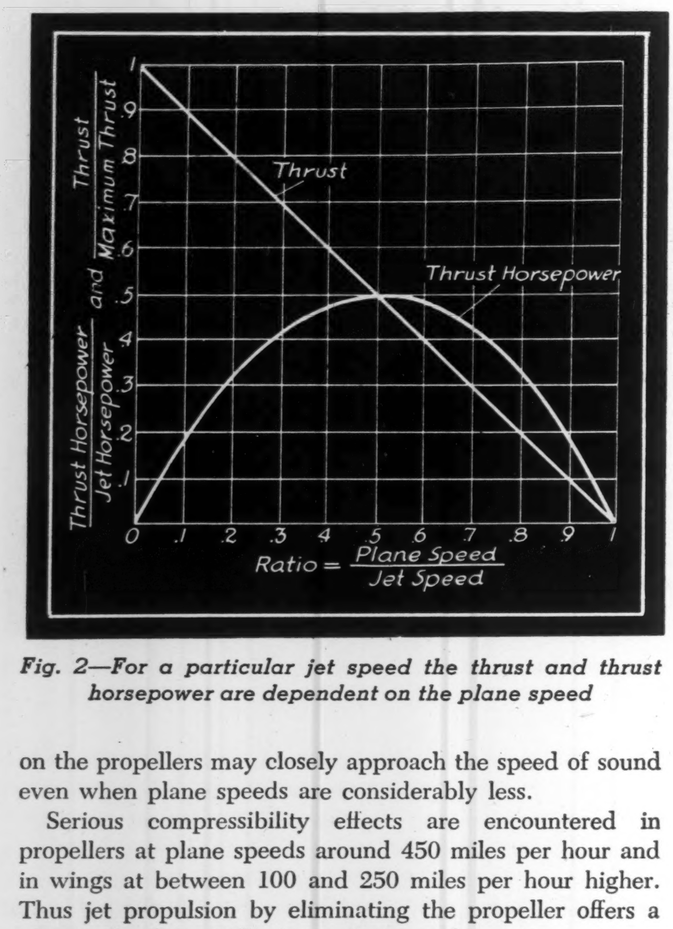


Fig. 2—For a particular jet speed the thrust and thrust horsepower are dependent on the plane speed

Serious compressibility effects are encountered in propellers at plane speeds around 450 miles per hour and in wings at between 100 and 250 miles per hour higher. Thus jet propulsion by eliminating the propeller offers a means of substantially increasing speeds.

How Thrust Is Generated

Propulsion of any type of airborne craft depends on the reaction from a jet. In the conventional plane the jet is furnished by the propeller, which projects a stream of air to the rear of the ship. In the J.P. plane the jet is created by burning fuel in air, causing it to expand, with the result that it issues from a nozzle at high velocity.

The diagrammatic sketch in *Fig.* 1 shows the principal features of a J.P. engine, which may be accommodated in a nacelle located in the wing. Air is drawn into the nacelle by a compressor which delivers it at a pressure in excess of atmospheric into the combustion chamber. Here a series of fuel jets spray oil into the air where it burns, furnishing heat to raise the air temperature. Expansion of the heated air and combustion products furnishes the power to operate a turbine for driving the compressor and to produce the high-velocity jet for propulsion of the plane. More fuel may be burned beyond the turbine to provide additional jet energy.

Thrust and horsepower furnished by a jet are readily calculated. The well-known equation force equals mass times acceleration ($F=ma$), may be rewritten force equals mass times rate of change of velocity ($F=mdv/dt$) or force equals rate of change of momentum. Assuming that the plane has forward velocity u feet per second while the speed of the jet (relative to the plane) is v feet per second, then the change of momentum of the air is $mv-mu=m\ (v-u)$. The rate of change of momentum is found by replacing m with the mass rate of flow of air in the jet, w/g, where w is the flow in pounds per second, giving the thrust equation

$$F=\frac{w}{g}(v-u) \ldots\ldots (1)$$

The thrust horsepower is proportional to the thrust force times the velocity of the plane, or

$$thp=\frac{Fu}{550}=\frac{w(v-u)u}{550g} \ldots\ldots (2)$$

Horsepower in the jet itself is proportional to the kinetic energy per second, or

$$jhp=\frac{w}{2g}\,\frac{v^2}{550} \ldots\ldots (3)$$

Ratio of thrust to jet horsepower is therefore

$$\frac{thp}{jhp}=\frac{2(v-u)u}{v^2}=2\left(1-\frac{u}{v}\right)\frac{u}{v} \ldots\ldots (4)$$

How this ratio varies with the ratio of plane speed to jet speed is shown in *Fig.* 2, from which it is evident that for a given jet (mass flow and velocity) the maximum thrust horsepower is developed when plane speed is one-half the jet speed, that is, when jet speed is twice the plane speed. The thrust, however, being directly proportional to the difference in speeds (Equation 1) is maximum when the plane is at rest, a fact which is of tremendous importance in takeoff. The high thrust at low plane speeds also is effective in providing fast climb. As the plane speed approaches the jet speed, thrust falls off to nothing.

Propulsion efficiency may be found by comparing the thrust horsepower with the energy that must be added to the air to produce the jet. Inasmuch as the air enters the propeller or J.P. unit with velocity u relative to the plane, the energy to be added is $\frac{1}{2}mv^2-\frac{1}{2}mu^2$ and the horsepower added is

$$php=\frac{w}{2g}\,\frac{v^2-u^2}{550} \ldots\ldots (5)$$

which may be called the propeller horsepower. The efficiency is therefore

$$e = \frac{thp}{php} = \frac{2(v-u)u}{v^2-u^2} = \frac{2\frac{u}{v}}{1+\frac{u}{v}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Variation of efficiency with ratio of plane speed to jet speed is shown in *Fig.* 3.

In conventional planes high efficiency is maintained by operating with a propeller jet speed only slightly greater than plane speed. Because the horsepower ratio is low a large jet is needed, this being attained through use of a large diameter propeller. Assuming a ratio $u/v = .75$, speed of the propeller jet at a plane speed of 450 miles per hour would be $450/.75 = 600$ miles per hour. Relative air speed of propeller blades and air, which is the vector sum of air speed and blade speed, will be even higher. Furthermore, inasmuch as lift or thrust on an airfoil section is caused primarily by pressure reduction on the back surface of the section due to increased air velocity over that surface, local air speeds at certain points may easily exceed the average by 30 per cent or more. On the basis of 600 miles per hour relative speed, local air speeds may therefore exceed $600 \times 1.3 = 780$ miles per hour, resulting in the so-called "compressibility burble" that occurs near the speed of sound.

At high altitudes the speed of sound, which is a function of temperature, becomes less than at sea level. For example, at 30,000 feet it is only 680 miles per hour, which means that the limitations imposed by compressibility burble are more severe than at sea level.

Jets Are Faster Than Speed of Sound

In the jet-propulsion plane the mechanism which furnishes the jet is not subject to limitations due to attaining the speed of sound, and jet speeds may considerably exceed this value. Speed of the jet is a function of the ratio of fuel to air in the J.P. unit, a high percentage of fuel per pound of air resulting in high jet velocity but low propulsive efficiency unless the plane speed also is high.

To show the type of calculations that might be involved in designing for jet propulsion the following hypothetical figures are offered, assuming a plane speed of 525 miles per hour or 770 feet per second. A probable jet velocity may be calculated by considering what happens to the air which enters the nacelle at the forward end, *Fig.* 1. It will be assumed that the pressure in the combustion chamber is to be three times atmospheric. Part of this pressure increase may be acquired from the arrest of the incoming air but most of it is provided by a rotary compressor. At sea level the dynamic pressure would not exceed $\rho u^2/2 = .00237 \times 770^2/2 = 700$ pounds per square foot or 4.87 pounds per square inch, ρ being the density of air at sea level, slugs per cubic foot. With compressor discharge pressure $3 \times 14.7 = 44.1$ pounds per square inch and intake pressure $14.7 + 4.87 = 19.57$, the pressure ratio is $44.1/19.47 = 2.26$. For this ratio and an air temperature of 50 degrees Fahr., the horsepower may be calculated from the information given in the article "How Superchargers Aid High-Altitude Flight", MACHINE DESIGN, August, 1943. Using Equation 2 and *Fig.* 4 from that article, the horsepower is found to be $.00573\ YT_1 = .00573 \times 133 = .762$ horsepower per pound of air per minute or 45.7 horsepower per pound of air per second. Assuming compressor efficiency 80 per cent, the horsepower absorbed will be $45.7/.8 = 57$, while the air will leave the compressor at a temperature of $50 + 133/.8 = 216$ degrees Fahr.

With enough fuel burned in the combustion chamber to raise the temperature of the mixture, to say, 1500 degrees Fahr., the total heat drop through the turbine and jet nozzle for the 3 to 1 pressure ratio will be 132 Btu per pound, from *Fig.* 8 in the article previously mentioned. Assuming turbine and jet efficiency 85 per cent, the available energy will be $132 \times .85 = 112.1$ Btu per pound per second or $112.1 \times 778/550 = 158$ horsepower. Of this, 57 horsepower will be absorbed by the compressor, leaving 101 horsepower at the jet. From Equation 3 the jet velocity would therefore be $(2 \times 32.2 \times 550 \times 101)^{1/2} = 1900$ feet per second. Hence for a flow of one pound per second the thrust, given by Equation 1, would be $(1900 - 770)/32.2 = 35.1$ pounds and the thrust horsepower $35.1 \times 770/550 = 49$. Propeller horsepower (Equation 5) would be $(1900^2 - 770^2)/(2 \times 32.2 \times 550) = 85.2$, giving a propulsion efficiency of somewhat less than 60 per cent, compared with better than 80 per cent for a good modern propeller.

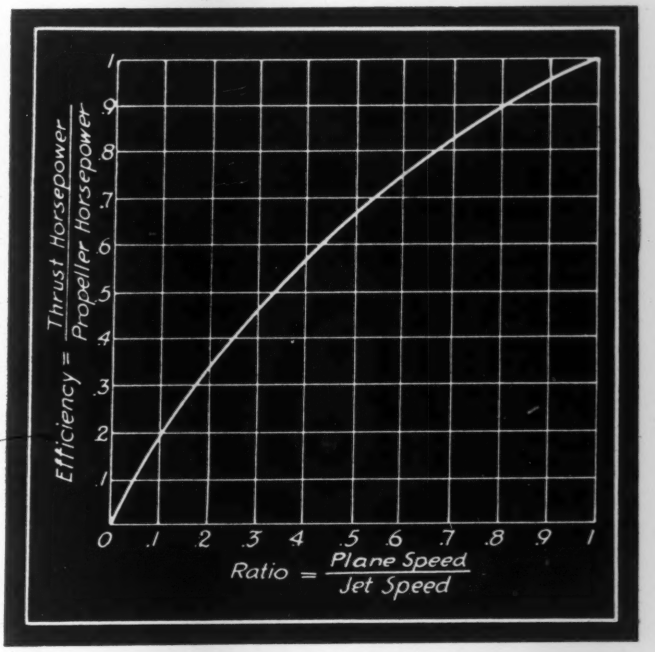


Fig. 3—Propulsion efficiency of a jet is highest when the plane speed is only slightly less than the jet speed

Translating the foregoing figures in terms of an actual plane, a fighter requiring, say, 2000 thrust horsepower at 525 miles per hour would use $2000/49 = 40.8$ pounds of air per second or 20.4 pounds per nacelle in a two-engine ship. The compressor plant would absorb no less than $40.8 \times 57 = 2330$ horsepower. This indicates the importance of the compressor and points to one of the reasons why the J.P. principle could not be applied many years ago. Hitherto rotary compressors have been of low efficiency and it is only since the development of a superior type of axial-flow compressor that both gas turbines and jet propulsion have become practically feasible.

Heat necessary to raise the air temperature from 216 degrees (at compressor discharge) to 1500 degrees (at turbine entrance) is approximately .26(1500 − 216) = 334 Btu per pound, where .26 is the specific heat of air at constant pressure for this temperature range. Using an oil such as kerosine with heating value 18,500 Btu per pound, the fuel rate would be 334/18,500 = .018 pounds per second. On the basis of propeller horsepower, this is a specific fuel consumption of .018 x 3600/85.2 = .76 pounds per horsepower hour, compared with less than .5 pounds per horsepower hour for a modern gasoline engine.

This high specific fuel consumption combined with low propulsion efficiency indicates that a jet-propelled plane working under the assumed conditions might be expected to use more than twice as much fuel as a conventional plane. This may be offset partly by the lighter weight of the J.P. engine, but shows that its use would be limited to short-range craft.

To improve the fuel consumption it would be necessary to increase the combustion temperature above the value of 1500 degrees Fahr. used in the calculations. While available blade materials set a limit to the temperature that can be used in the turbine, higher temperatures can safely be used in the nozzle and jet. Consequently a portion of the fuel can be burned in the main combustion chamber and the remainder in the passage beyond between the turbines and the nozzle. This adds to the energy in the jet and furnishes a higher jet speed which more than offsets the additional fuel that must be consumed. But for the same plane speed a higher jet speed means lower propulsive efficiency which partly offsets the gain in thermal efficiency. The conclusion is that only high-speed planes which can use a high-speed jet without unduly low propulsive efficiency can utilize jet propulsion effectively.

Does Not Require High-Test Fuel

Advantages which justify its use in spite of high fuel consumption are, in addition to the higher speeds possible, the cheaper, less volatile fuel that can be used, the absence of vibration, and the difficulty of detection by the enemy due to the absence of propeller and exhaust noises as well as to the elimination of electrical ignition.

At high altitudes jet propulsion comes into its own. Although the jet plane, unlike the rocket, depends on atmospheric air and the power tends to fall off with decreasing density, it is much easier to provide the necessary air capacity to compensate for low density than it is in the case of a conventional engine. Hence high power is available at high altitudes while the resistance of the plane falls off, thus enabling the attainment of even higher plane speeds. Thermal efficiency also improves somewhat, consequently there is an all-around improvement in economy as well as in speed. Performance of the propeller plane, on the other hand, is limited by the fact that the propeller must work in atmospheric air, which at 30,000 feet has a density only 37 per cent of that at sea level. The J.P. unit has the advantage of being able to compress the air on which it works.

While many refinements have been ignored in the foregoing calculation, it is hoped that the method of approach will furnish some insight into the principles of jet propulsion.

Designing Against Carelessness

THREE underlying fundamentals of aircraft engineering that must influence every design decision are: Safety, weight, and serviceability. The great importance of these factors has caused the aircraft industry to set up and staff enormous engineering departments to do a detailed engineering job with a thoroughness not required in any other industry. Safety of present aircraft is due to the continuous vigilance and efforts of the government agencies, the airlines, and manufacturers, and the designer must not relax or these past gains will be lost.

Many people seem to feel that airplanes crash only when some major catastrophe occurs. Even a cursory study of accident records will disabuse one quickly of this opinion. Airplanes have crashed because the pilot dropped a microphone, or because he happened to kick a small lever in the floor. Several accidents have been due to the malfunctioning of comparatively minor pieces of accessory equipment.

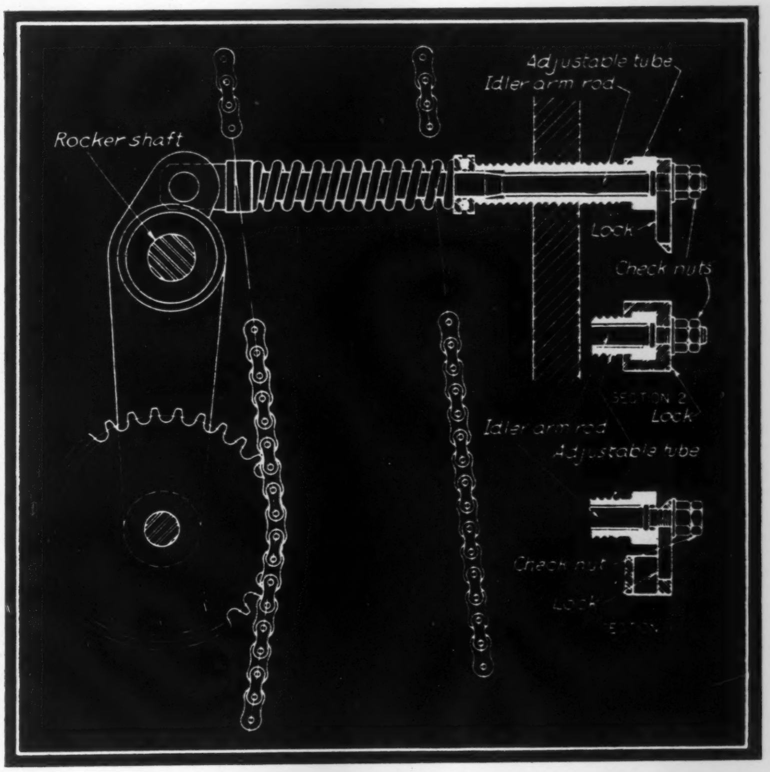
Treated with Care or Kicked Around?

The young engineer is likely to feel that the device he is designing is so fine and wonderful that it deserves the utmost care and consideration by the flight crew, the ground personnel and even by the enemy. He is terribly shocked to discover that his brain child is kicked around and given the same type of care as are tools supplied with second-hand automobiles. The sound and only safe attitude to take in the development of any gadget for an airplane is to assume that it will be neglected, lubricated with the wrong oil, maladjusted, and misused. If the device will still function under these circumstances it may be satisfactory; if it does not, it will be potentially dangerous. As every device and piece of equipment is subject to failure, the results of such failures must be considered. If these failures cannot cause accidents, then the device may be satisfactory.

Failure of maintenance personnel to replace parts may be as much an engineering failure as are those due to poor design. In many cases designers have made parts so that it is impossible for inspection personnel to determine if these parts are properly installed. In such cases failures due to the omission of necessary bushings, bolts, etc., can properly be blamed upon the carelessness of the designer. The designer must make every effort to be sure that there is only one possible method for assembling the component, and that the proper one. Assembly and adjusting sequences which involve procedures in different locations in the airplane are invariably dangerous since it is easy for some items to be left unattached or improperly adjusted.—*From a paper by A. L. Klein, Douglas Aircraft Co. Inc., presented at the recent national aeronautic meeting of the Society of Automotive Engineers.*

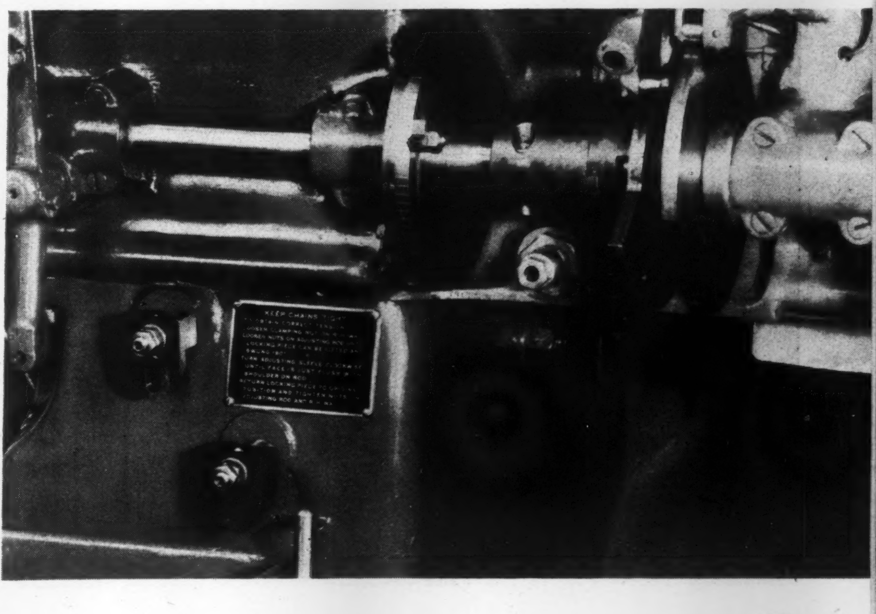
Scanning the Field for Ideas

Chain tension must be capable of accurate adjustment if smooth operation and long life are to be obtained from high-speed chain drives. Many automatic or self-compensating devices have been developed in the past but these have been unsatisfactory in that they either produced too much tension, causing chain wear, or failed to operate in service after stretch had taken place, allowing whip. A simple and effective method for accurate chain adjustment, however, has been developed by Brown & Sharpe for use on their automatic screw machines which employ chain speeds of 5000 feet per minute. The unit is so designed that excessive pressure cannot be employed.

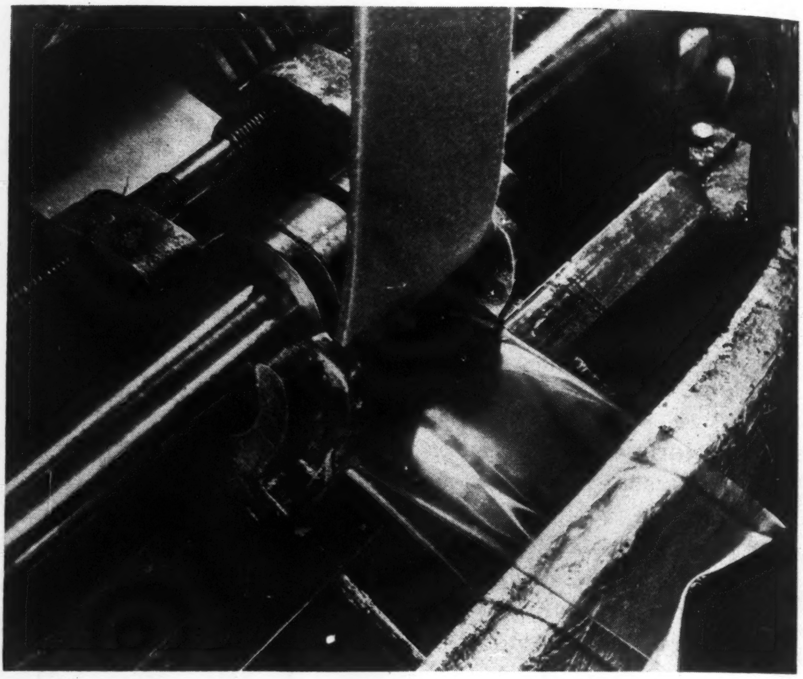


Principle of the device is shown in the drawing at right and the location of the adjusting nuts in the photograph below. For adjustment the check nuts are backed off allowing disengagement of the lock from the hex head on the adjusting tube. This position is shown in Section 1 in which the rod has moved as far as the spring can push it. The tube then may be screwed in until its end is flush with a shoulder on the rod. At this point the spring has an exact compressed length and the desired pounds of force are acting. A ball thrust bearing between the tube and spring reduces torque on the tube and permits it to turn without twisting the spring.

Lock may then be dropped over the hex head of the tube and the check nuts turned up tight as shown in Section 2. Should the tube have been screwed in too far, giving an excessive spring pressure, the lock would not drop into position because its slot would not drop over the shoulder on the rod. In such a case resetting would be necessary. As a final clamping means a nut on the rocker shaft is



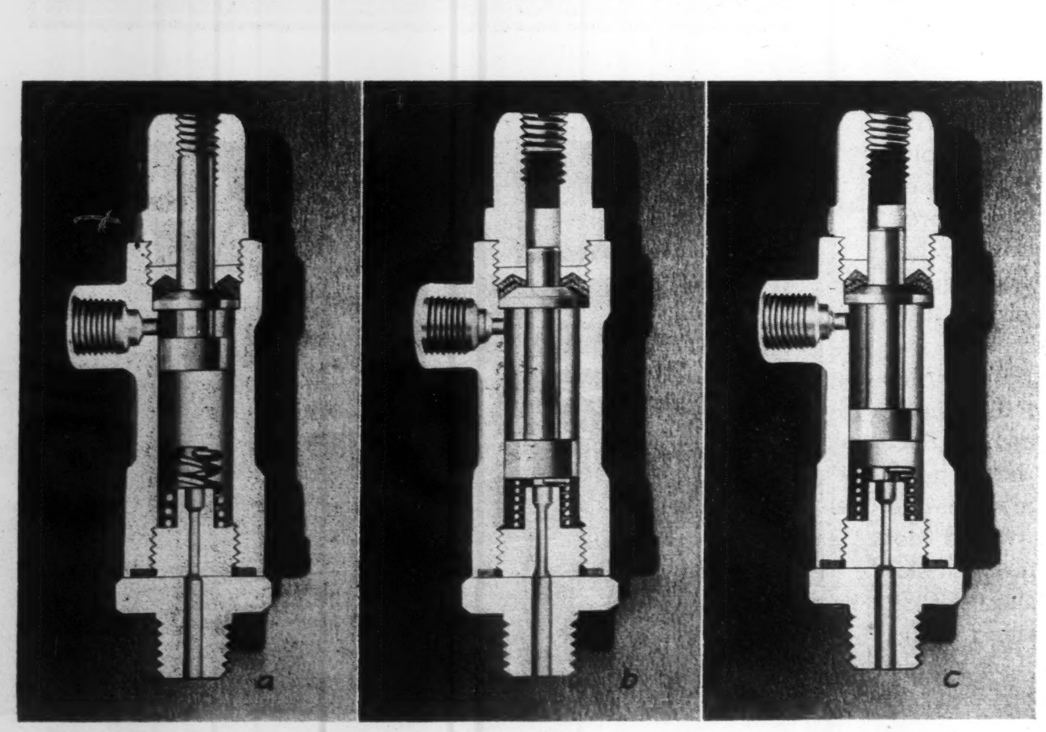
tightened, causing the idler arm to be pinched between a thrust collar on the shaft and a thrust face on the machine bed. Thus an effective tensioning device is obtained to assure proper adjustment.



Two-ply leather belting in which the plys are bonded with plastic foil at normal belt-drive tension has minimized some of the characteristic weaknesses of such belting. Produced by a continuous process, Celluloid foil passes through a solvent bath and is fed between the two plys of leather which pass between sets of rubber pressure rolls, shown at right. The laminate is then wound on a drying drum while under normal tension, resulting in a uniformly bonded belt with high adhesive strength far superior to hand cementing methods previously employed. This double-ply belt, developed by Graton & Knight, is flexible and has a high coefficient of friction and a minimum of stretch.

Differential pressures actuate metering valves for a centralized lubrication system, developed by Alemite, allowing the use of a single hydraulic line with the valves connected in parallel to the line. In this system the valve operation is as indicated in the sectional views below. At a, the valve is full of lubricant ready to discharge the measured quantity of oil to its bearing through the passage at the bottom. This quantity is controlled by a screw setting above the piston stem.

To deliver oil, pressure is admitted through the side port forcing the piston down and the measured oil into the bearing. Continued pressure moves the piston against a spring-loaded check valve, closing the bottom port as shown at b. As the pressure continues to build up, leakage past the piston causes it to begin to rise as at c while holding the bottom port closed, both due to difference in pressure. When the piston moves to the top, pressure builds up to the shut-off valve of the pump indicating that all valves have operated. As the pressure subsides the spring-loaded check valve again opens which returns the system to the position shown at a.



Tailor-Made Controls Utilize Standard Component Parts

By D. K. Frost
Electrical Engineer
Mattison Machine Works

DURING the development of high-production, multimotored machines in the past ten years, a gradual evolution in the electrical controls has taken place. This evolution has not been completed, but enough progress has been made to define the general trend in design. Some machines lend themselves readily to the use of built-in control, other types make it imperative to mount the control in a separate cabinet. This cabinet, if possible, should be mounted on and made a part of the machine so as to make it self-contained. The machine can then be wired completely for testing and can be shipped without materially disturbing the wiring. It is certain that users prefer this method of procedure so that all they have to do when the machine is received is to attach the necessary line wires to make the machine ready to run.

In either case, whether the control is built in or contained in a separate cabinet, the general trend is toward tailor-made or built-up control panels, utilizing standard component electric control parts. The reasons for this are obvious when given proper consideration.

Changes Are Facilitated

The first and most compelling reason is that it is easier to make changes or additions to the original layout. For instance, the sales department may want to add a feature to the machine which requires a relay or perhaps two relays, or else they want to use a motor of higher capacity which means a larger switch. Or they may wish to add an electrically operated lubricating pump or a time-delay device. These all require changes on the panel. We constantly expect and try to foresee such requests by having space available whereby we can regroup the control items, make up a new drilling templet,

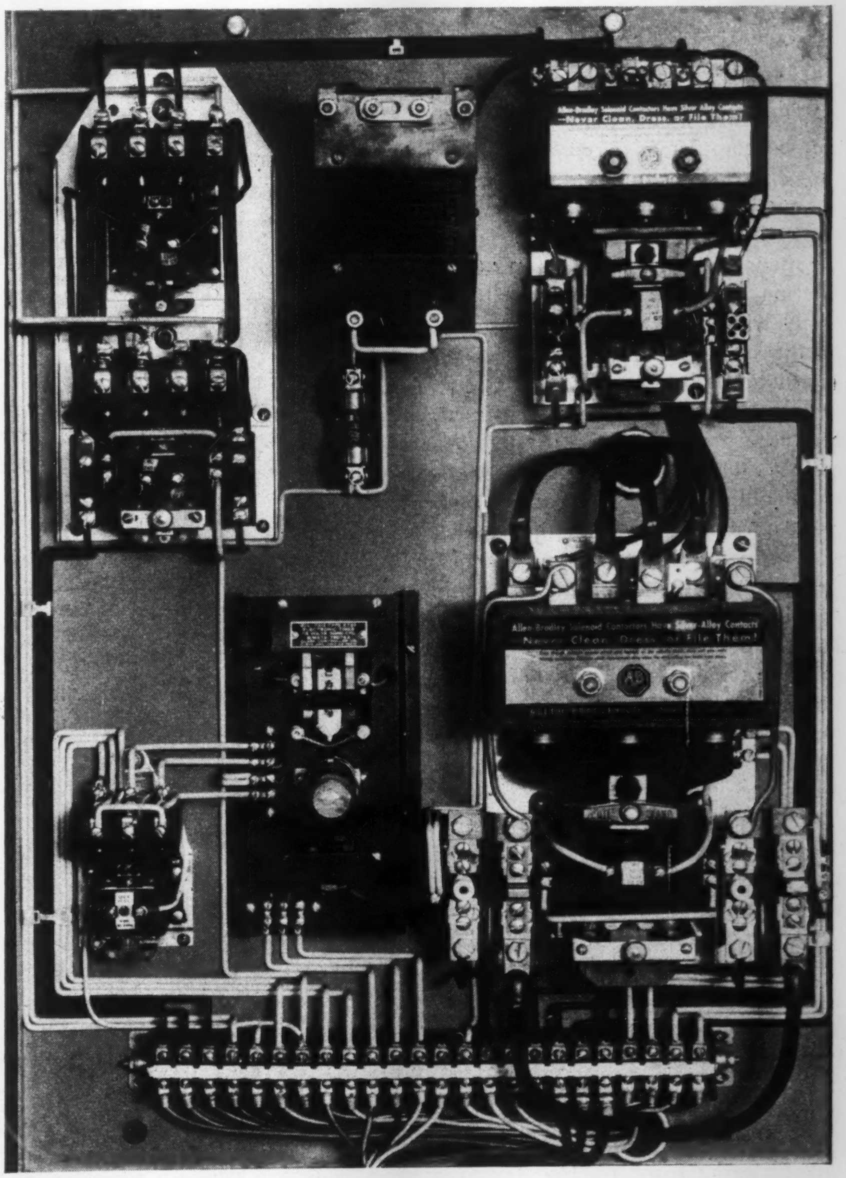

Allen-Bradley Solenoid Contactors Have Silver Alloy Contacts
—Never Clean, Dress, or File Them!
Allen-Bradley Solenoid Contactors Have Silver Alloy Contacts
Never Clean, Dress, or File Them!
AB

Fig. 1—Main panel for a large surface grinder, tailor made from standard component electrical parts

and take care of the desired changes. These alterations do not occur "once in a while"—they must be expected at any time and are a continuing process. The making of required changes on panels by the control manufacturer is not, from our own experience, a feasible proposition.

A second reason which is really an off-shoot of the first is that, during the excessive demand for equipment in 1942 and 1943, the shipping estimate on panels purchased from any supplier ranged from 16 to 24 weeks while it was still possible to get individual control items in six to eight weeks. As a matter of fact, by building our own panels and keeping a respectable stock of panel material and switches, we were able to have a stock of 12 to 15 panels ahead of the assembly floor at all times.

Built on Subassembly Panels

These control panels are built up on subassembly metal plates so that they can be completely wired and tested before the final assembly. At the height of our production schedule, two men working part time were required to do the panel wiring; but, in ordinary production, one man working full time can easily wire the panels and keep a check on the stock of parts required for future production.

A careful comparison reveals that assembled panels purchased in lots of twenty, which carry the maximum discount obtainable, cost on an average 35 per cent more than made-up panels. As our company has made up panels for practically all the machines manufactured in the past eight to ten years, we have had a chance to make such comparison over a long period of time and under varying conditions. While the item of cost is not a compelling reason, it is something to consider in the overall picture.

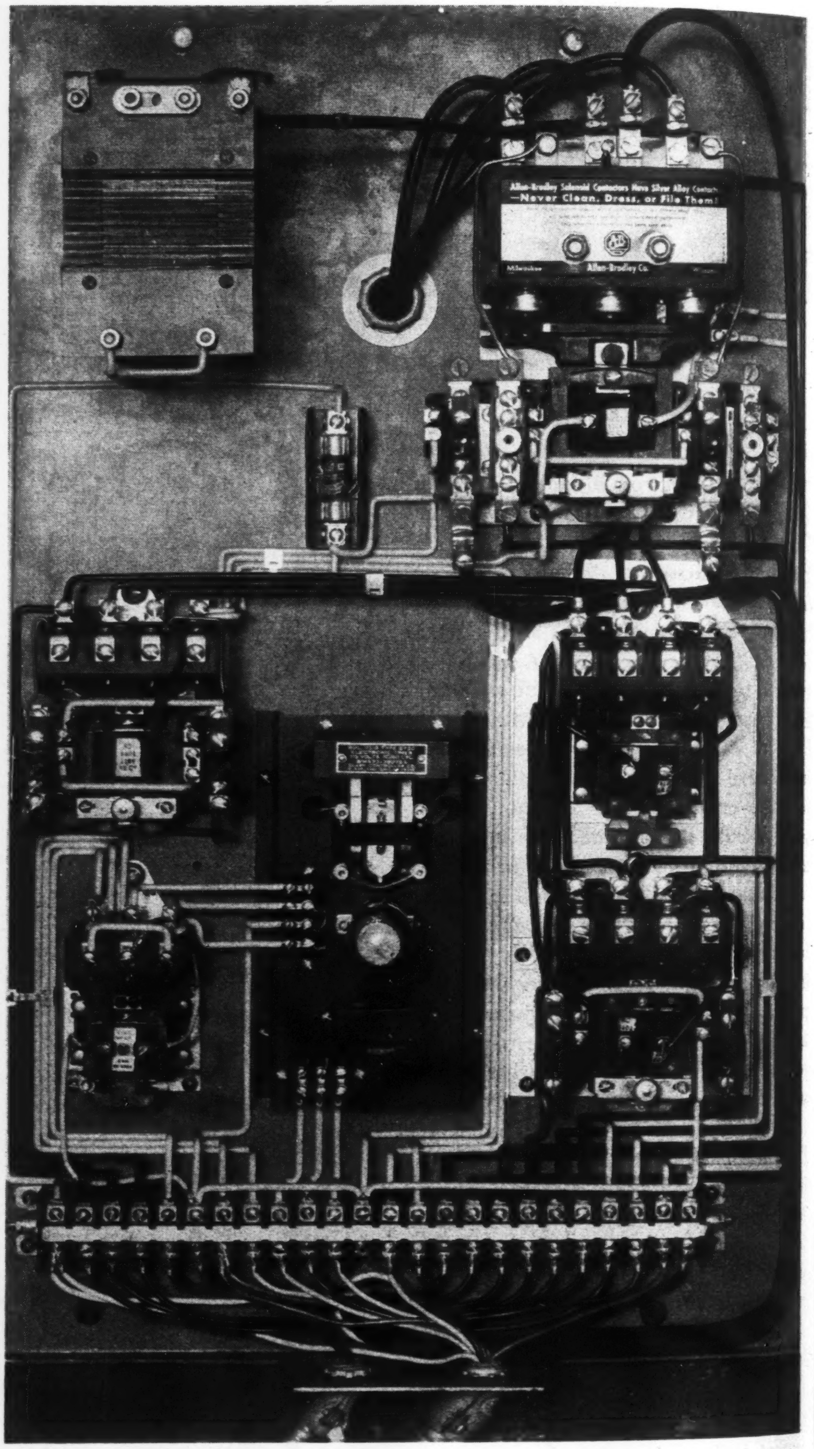

Allen-Bradley Solenoid Contactors Have Silver Alloy Contacts
—Never Clean, Dress, or File Them!
Allen-Bradley Co.

Fig. 2—Tailor-made panel for a small-size surface grinder

Referring to the panels shown in *Figs.* 1 and 2 it will be noticed that the control transformers have link connections at the top so that the primary may be 220 or 440 volts at either 60 or 50 cycles with the secondary 110 volts. Oversize switches are employed for all motors so that the panels can be used safely on either 220 or 440 volts and changed over in less than 15 minutes by changing the taps on the transformer primary and changing the overload heater elements on the individual switches.

Control circuit for the coils always remains at 110 volts. Being fused on the secondary side, short circuits have been prevented from doing untold damage to the overload relay and control contacts, not only because of the fuse protection but also because the power behind the short circuit is limited to the secondary capacity of a small insulated transformer, usually only 300 to 750 watts. This has been a fine thing for the war effort in saving many replacements because a short in the control with all the power of a shop circuit behind it can do a lot of damage in a few seconds.

Manufacturers of magnetic switches have always cooperated with machine manufacturers who make their own control panels by furnishing deadback switches which can be mounted safely on metal base plates. They have also made changes which provide more convenient connections for front-of-board wiring. Arrangements of line leads and motor connections as well as control leads are in general easily accessible and convenient for panel wiring.

Most switches are furnished with either normally open or normally closed electrical interlocks. Electric parts manufacturers have probably benefited as much from the demands for such features as have machinery manufacturers. Any change which makes it more convenient to produce tailor-made panels will make it that much easier to wire commercially assembled panels.

Need for Standard Mounting

Standardized mounting dimensions for size 1, 2, and 3 magnetic switches have been needed for some time. In reality the physical dimensions are so close now, it is amazing that three holes are not located so that the mounting dimensions at least may be standard. In this connection it is not necessary that all switches be the same size but merely that the mounting-hole dimensions be the same on each switch. This would be an immeasurable benefit in providing interchangeable unit mounting. We don't worry too much about the size as we generally have space available to accommodate at least the next size larger switch. It is hoped that some day standard mounting dimensions will be a fact. When it is achieved, the universal question will be, "Why wasn't it done sooner?"

A tailor-made panel is shown in *Fig.* 3 mounted on the column of a machine. Because electronic devices may be used on this panel, it has been found desirable to make provision for circulation of cooling air. The particular electronic device used in this instance, however, does not need cooling.

The panel in *Fig.* 4 has a front-operated disconnect switch and is a type 1A panel with felt gaskets to make the panel dustproof, as required by some users.

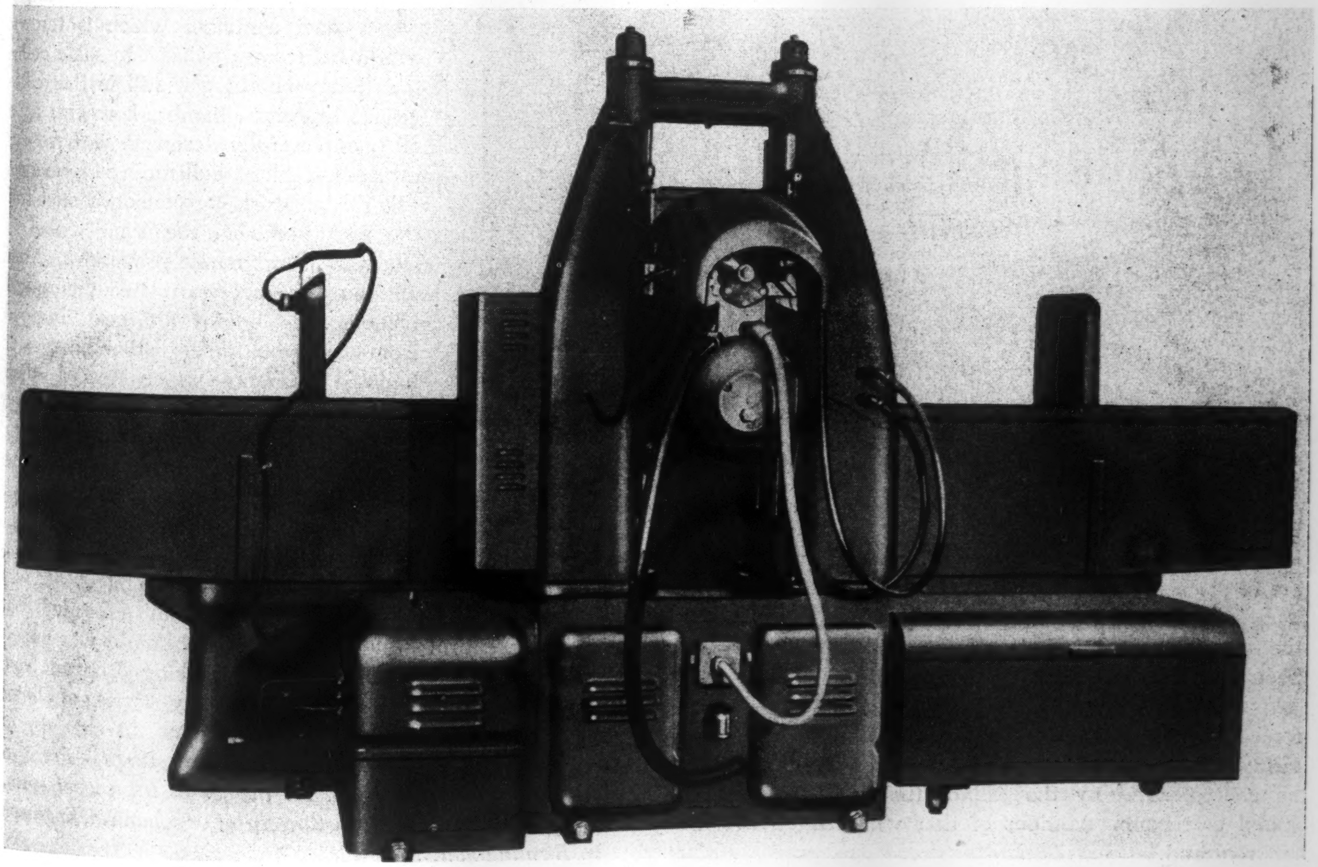
Enclosures for control constitute an open question. Undoubtedly, there are advantages in having dustproof enclosures. The general term dustproof means apparatus so constructed or protected that the accumulation of dust will not interfere with its successful operation. Now that term is subject to a broad interpretation. If, over a period of time, panels with vents for air do not allow an accumulation of dust that interferes with successful operation, we are permitted to say that such apparatus is dustproof.

On the other hand, if apparatus is specified as dust-tight that term is not subject to broad interpretation because dust-tight means that the apparatus is so constructed that dust cannot enter the enclosing case. The same interpretations are permitted with waterproof and watertight apparatus.

The panel shown on the machine in *Fig.* 3 is not only dustproof but is also splashproof since the openings are not located on the side which is subject to splash from the coolant used in the grinding operation. No trouble or trace of trouble has occurred over a long period of successful operation.

We have in all panel and machine wiring followed as closely as possible the recommendations of the Machine Tool Electrical Standards. This code was adopted in 1942 by the American Standards Association as an Amer-

Fig. 3—Grinder showing panel of Fig. 2 mounted on column



ican War Standard. Certain parts of the code have been covered in substance in Article 670, "Machine Tool Wiring", in a supplement to the National Electric Code.

The code as written is constructive in detail and, if followed, should insure a good workmanlike wiring job.

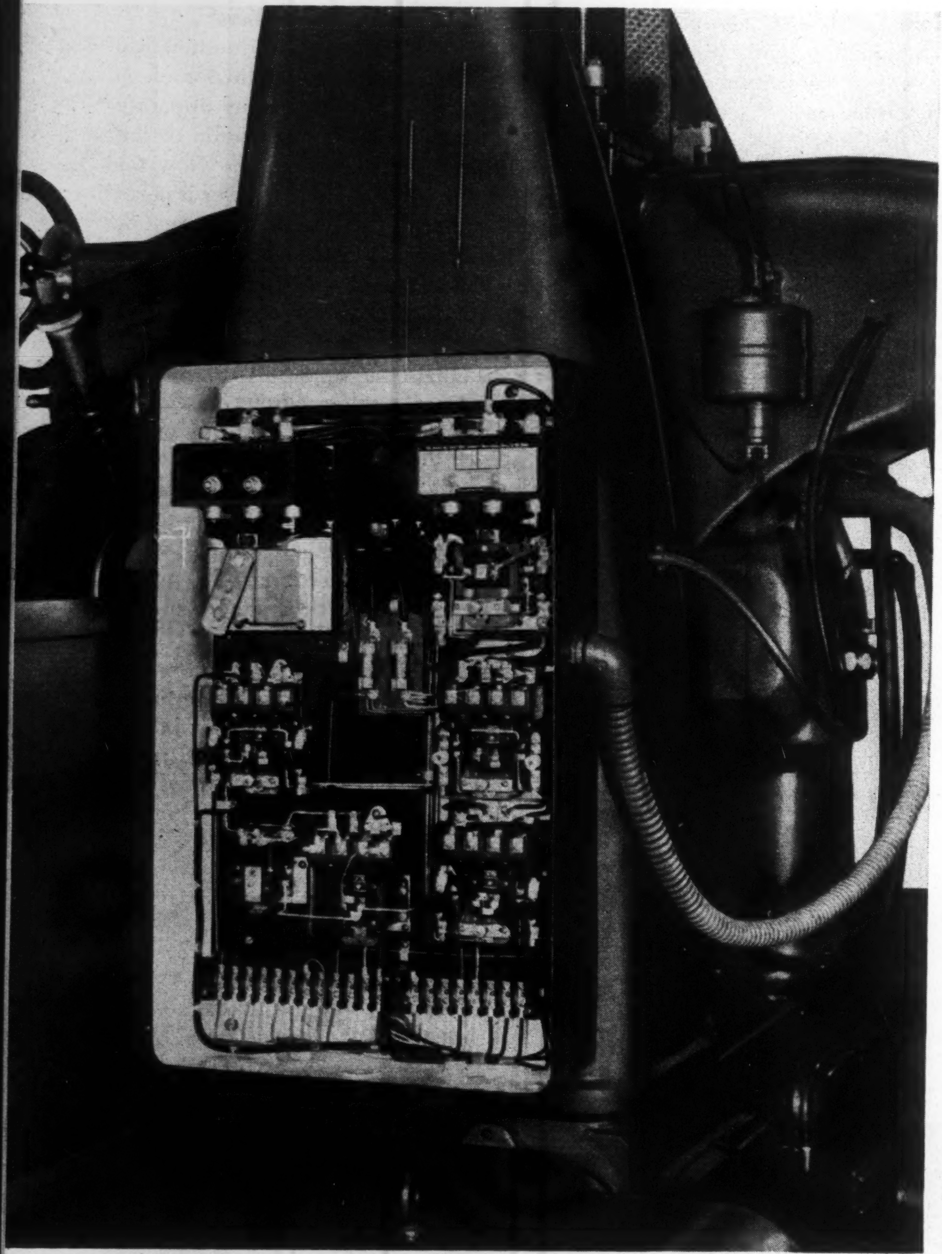


Fig. 4—Cover removed from cabinet showing arrangement of control devices and wiring for a precision grinder

It is certain that the code is not responsible for many of the criticisms that one hears about various machines. It is equally certain that if all manufacturers had followed the standard code, much of the criticisms would have been avoided.

No doubt the code can be improved and be made more forceful by making vital articles mandatory as characterized by the word "shall". Advisory rules would then be characterized by the use of the word "should", or stated as recommendations of that which is advised but not required.

Representatives of the General Motors Corp. have made some specific recommendations as to changes which should be incorporated. Most of these recommendations are exceedingly constructive. It is to be expected that some of the requirements will cause changes but this is all to the good, because evidently changes are in order not only to expedite the standardization of hydraulic and electrical systems but also to achieve other desirable features.

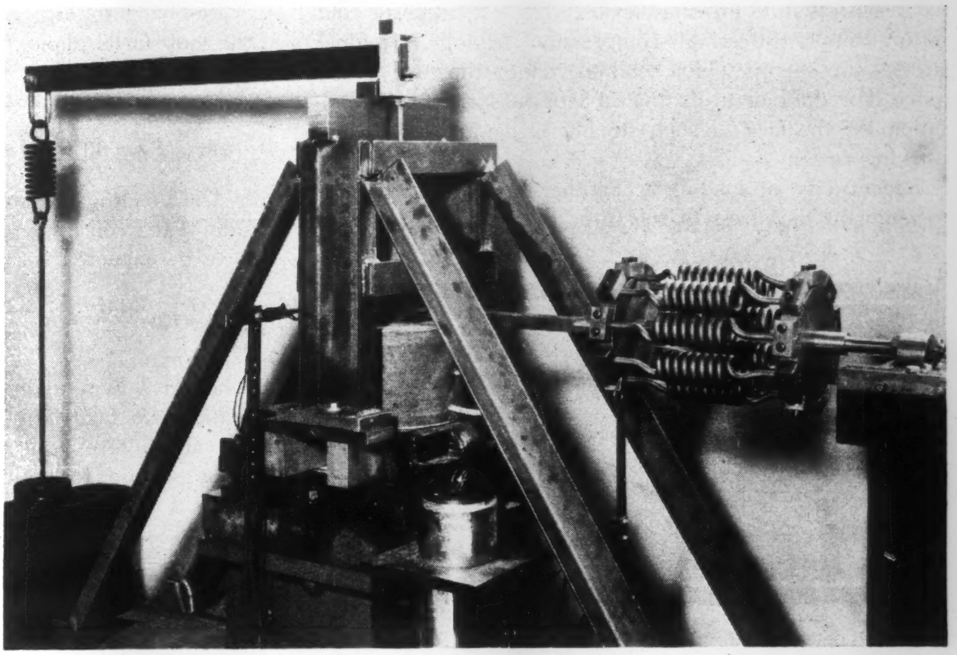
Achieved Worthwhile Results

It is always refreshing to have large users of machines make suggestions on how to improve them. Engineers of machinery manufacturing companies have, I believe, been instrumental in presenting a clear picture of actual requirements to the producers of electrical equipment and worthwhile results have been achieved. The actual users of machines, however, are in a still better position to present the final factual picture of their needs. The closer these various groups can get to each other's viewpoints and requirements, that much sooner will the final answer be nearer to perfection.

Precision Design

TO MANY people a bomb is simply a steel container which is thrown overboard from a plane. In some countries today bombing is still in that category. American bombs, however, have all been carefully designed, as have our projectiles, for ballistic performance. The design starts by placing a model of the new bomb in the wind tunnel to study its aerodynamic properties. After obtaining the necessary streamlining and balance, the bombs are next dropped from various altitudes under the eyes of ballistic cameras which record their movement through the sky. Aberdeen Proving Ground, where these ballistic measurements are made, prepares the ballistic tables used for the graduation of the precision bombing sights.

When bombs were first employed, the altitudes at which planes could carry them were decidedly limited. It was found that, when the velocity of the bomb exceeded that of sound, considerable redesign of our bombs was necessary and they are now capable of being dropped from the highest altitudes. Today, many types of bomb fuses are required to burst the bomb in the air, on or under the ground, and at different times.—*From an address by Maj. Gen. G. M. Barnes at the recent spring meeting of the American Society of Mechanical Engineers in Birmingham, Ala.*



Tension-compression springs in fatigue testing machine have ends brought out in gradual loop, coinciding with spring axis

How End-Coil Design Affects Tension Springs

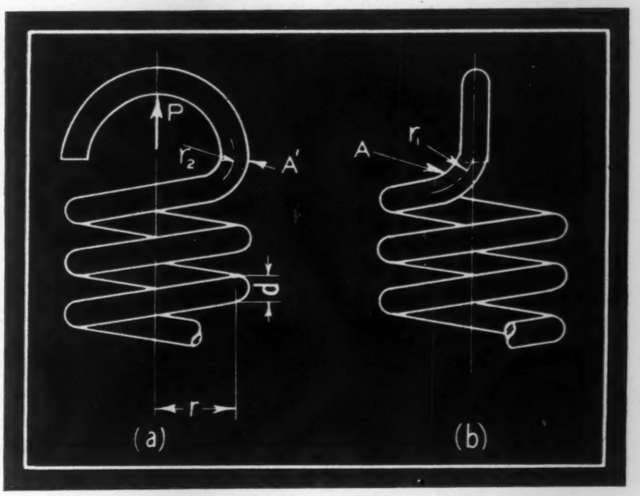
By A. M. Wahl
Westinghouse Research Laboratories

IT IS customary practice in spring design to use somewhat lower working stresses for helical tension springs than those permissible for compression springs of the same material and wire size. The reasons why this is necesary are somewhat obscure. Analysis indicates, however, that the practice is justified when all factors are considered. The present article will discuss these from the standpoint of fundamentals of strength of materials (1)*.

In the usual calculation of helical tension springs it is assumed that the load is applied axially, stresses being calculated in the body of the spring in the same manner as is done for compression springs. However, in forming the end loops the curvature at points such as A and A' (*Fig.* 1) may be considerably greater than the curvature of the bar in the body of the spring. This results in additional stress concentration effects which may be high, particularly if the ends are carelessly made. As a result, failure of such springs usually occurs in the bend at points such as A or A', and the spring is materially weaker than a similar compression spring. It should also be noted that, as will be shown later, even if the end loop is made with a radius of curvature no smaller than that of the coil, some weakening effect is indicated. To reduce this weakening effect to a minimum the least radius of curvature of the end loops should be kept as large as practicable.

A further reason why tension springs should be stressed lower than compression springs lies in the fact that it is difficult to cold set such springs properly without losing initial tension or obtaining excessive space between turns.

Fig. 1—Tension spring with sharp curvature at end loop



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*Numbers in parentheses refer to references at end of article.

As discussed in a previous article (2), by properly cold setting or prestressing compression springs, residual stresses are set up which tend to reduce the load stress. Since it is difficult to do this in tension springs, compensation for the lack of such stresses is made by reducing working stress.

Eccentricity of loading is another factor which tends to augment the stress in the turns of tension springs but this also occurs in compression springs (2). Because of variations in the hook or in the method of attachment of the spring in practice, the resultant load P in *Fig.* 1 may be displaced from the axis by a small amount. This results in an increased stress which may require consideration. Where such springs are highly stressed it is desirable to keep this eccentricity to a minimum.

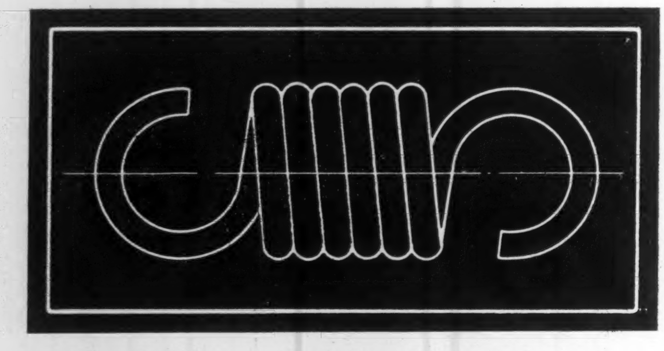

Fig. 2—Spring with full end loop turned up

Exact Stress Determination Is Difficult

End Loop Turned Up at 60 Degrees: In the case of a tension spring with a full end loop turned up as indicated in *Fig.* 2, because of the complicated mathematical shape of the space curve representing the loop an exact analysis of stress is difficult. Also the exact shape of the space curve will vary among different springs and for this reason an analysis applying to a given spring would not necessarily apply in other cases. For an end loop formed by bending up a half coil as indicated in *Fig.* 1 the analysis also is complicated. However, the special shape of loop shown in *Fig.* 3, which consists of a loop turned up at an angle of 60 degrees (*Fig.* 3*a*) can be analyzed approximately without difficulty. For this reason this analysis will be carried out primarily to throw some light on the stress conditions in similar end turns.

For a loop turned up at 60 degrees as shown in *Fig.* 3 the radius of curvature at all points is equal to the coil radius r. Since there are no sharp bends in such a loop, it represents a more favorable condition from the standpoint of peak stress than the loop of *Fig.* 1 which has a sharp bend at the point where it joins the body of the spring.

Considering Stress Due to Curvature

Analysis Including Curvature Effects: In carrying out this analysis the stress augment due to curvature of the wire will first be considered. Later a similar analysis neglecting curvature effects (applicable where static loading is involved) will be treated.

Referring to *Fig.* 4*a* which represents a plan view of the loop in its plane, the end force acting in the plane of the loop will be $P \sin 60°$. Thus the bending moment on section a-a' at an angle θ will be

$$M_b = P\sin 60°(r\sin\theta) \qquad (1)$$

The bending stress σ_b due to the moment M_b at the inside of the coil at a' (*Fig.* 4*a*) where the maximum stress is to be expected is

$$\sigma_b = K_1\frac{32M_b}{\pi d^3} = K_1\frac{32Pr\sin 60°\sin\theta}{\pi d^3} \qquad (2)$$

where d = wire diameter, r = mean coil radius and K_1 = a factor to take into account the stress augment due to curvature. An approximate expression for K_1 for a circular section for values of spring index c about 3 or more is (3)

$$K_1=\frac{4c-1}{4c-4} \qquad (3)$$

Due to the load component $P \sin 60°$ (*Fig.* 4*a*) there will be a direct tension stress σ_t acting on section a-a'. This stress σ_t is equal to load divided by area or

$$\sigma_t=\frac{4P}{\pi d^2}\sin 60°\sin\theta \qquad (4)$$

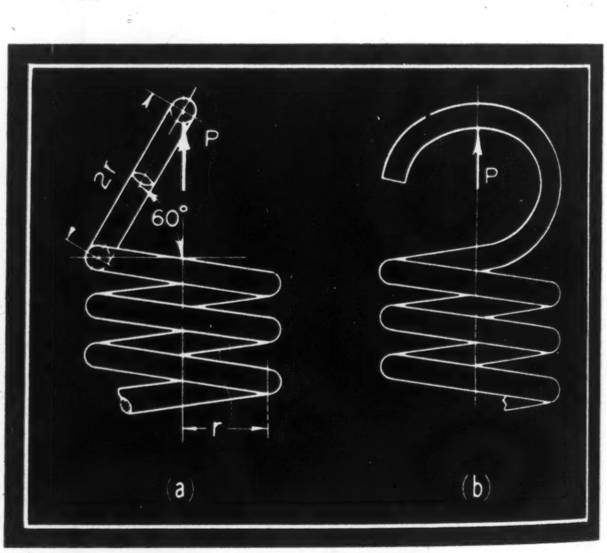

Fig. 3—Full loop turned up at 60-degree angle

The peak stress σ at a' will be the sum of the bending stress σ_b and the tension stress σ_t. Adding these stresses as given by Equations 2 and 4 and simplifying,

$$\sigma=\sigma_b+\sigma_t=\frac{32Pr\sin 60°}{\pi d^3}\left(K_1+\frac{1}{4c}\right)\sin\theta \qquad (5)$$

At section a-a' (*Fig.* 4*a*), a torsion moment M_t will act due to the load component $P \cos 60°$ (*Fig.* 4*b*) acting perpendicular to the plane of the loop. This torsion moment is

$$M_t=Pr\cos 60°(1-\cos\theta) \qquad (6)$$

Assuming that the stress augment due to curvature will be approximately the same in torsion as in bending (which is borne out by tests) the factor K_1 may be used to calculate the shear stress τ_1 due to the twisting moment M_t. This gives

$$\tau_1=K_1\frac{16M_t}{\pi d^3}=K_1\frac{16Pr\cos 60^\circ(1-\cos\theta)}{\pi d^3} \qquad (7)$$

The load component $P\cos 60^\circ$ perpendicular to the plane of the ring (*Fig. 4b*) will produce a direct shear stress τ_2 at a' (*Fig. 4a*). This shear stress may be taken equal to that in a round cantilever acted on by an end load $P\sin 60^\circ$, which is (4)

$$\tau_2=1.23\left(\frac{4P\cos 60^\circ}{\pi d^2}\right) \qquad (8)$$

Adding the stresses τ_1 and τ_2 as given by Equations 7 and 8, and simplifying, the shear stress τ at point a', *Fig. 4a*, becomes

$$\tau=\tau_1+\tau_2=\frac{16Pr\cos 60^\circ}{\pi d^3}\left[K_1(1-\cos\theta)+\frac{.615}{c}\right] \qquad (9)$$

The bending stress σ and shearing stress τ acting on the inside of the coil at a' must be combined according to a *strength theory* to get the equivalent shearing stress.

DISCUSSING some of the important factors in the design of tension springs, this article is based on a chapter in the author's book "Mechanical Springs" which will be published this month. In this article particular attention is given to stresses in end coils as well as to initial tension and working stress

Since in recent years there has been an increasing tendency to use the *shear-energy theory* as a basis for calculation this theory will be used in the following. On this basis the equivalent shearing stress τ_e becomes (5)

$$\tau_e=\sqrt{\tau^2+\tfrac{1}{3}\sigma^2}$$

Using Equations 5 and 9 in this the stress τ_e becomes, after simplification,

$$\tau_e=\tau_n\sqrt{\frac{1}{4}\left(1-\frac{K_1\cos\theta}{K}\right)^2+\left(\frac{K_2\sin\theta}{K}\right)^2} \qquad (10)$$

where

$$\tau_n=K\frac{16Pr}{\pi d^3} \qquad (10a)$$

The stress τ_n is that figured in the usual way including curvature.

$$K=\frac{4c-1}{4c-4}+\frac{.615}{c} \qquad (11)$$

$$K_2=K_1+\frac{1}{4c} \qquad (12)$$

Equation 10 gives equivalent stress τ_e as a function of θ. To find the angle θ_m where τ_e has its maximum value, Equation 10 must be differentiated with respect to θ, and the result set equal to zero. This gives

$$\cos\theta_m=\frac{KK_1}{K_1^2-4K_2^2} \qquad (13)$$

Thus for any given index c, the values K, K_1, and K_2 may be found and from these $\cos\theta_m$ is calculated using Equation 13. This value of θ_m is used in Equation 10 to find the maximum equivalent stress $(\tau_e)_{max}$. The final result becomes

$$(\tau_e)_{max}=C_1\tau_n \qquad (14)$$

where

$$C_1=\sqrt{\frac{1}{4}\left(1-\frac{K_1}{K}\cos\theta_m\right)^2+\left(\frac{K_2\sin\theta_m}{K}\right)^2} \qquad (15)$$

Since τ_n is the stress in the body of the spring figured in the usual way including curvature effects, the factor C_1 represents a stress-raising or stress-concentration effect due to the end turns.

TABLE I

Factor C_1 as Function of Spring Index

Index c	Factor C_1
3	1.08
5	1.10
10	1.125
∞	1.155

Calculated values of C_1 for various values of spring index c are given in TABLE I.

The table indicates that for most practical springs which have indexes between 5 and 10 the equivalent stress in the 60-degree type of end loop shown in *Fig. 3* will be around 10 to 13 per cent higher than the stress in the body of the spring. This is true even though the radius of curvature of the end loop is no smaller than that in the body of the spring. For an end loop with a

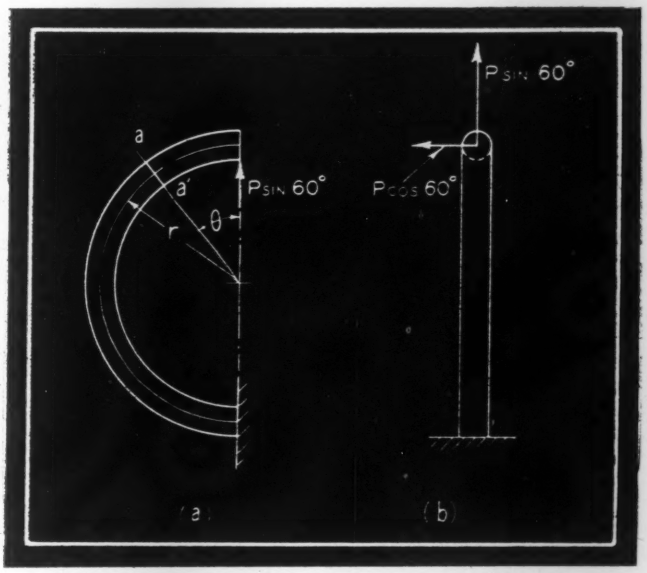

Fig. 4—Forces acting on end loop of Fig. 3

sharp bend as shown in *Fig.* 1, an even greater weakening or stress-concentration effect may be expected. For the full loop turned up as shown in *Fig.* 2 conditions are probably somewhat similar to the 60-degree type of loop shown in *Fig.* 3.

This analysis thus indicates that some stress-concentration effect due to the end turns may be expected even when the minimum radius of curvature of the end loop is kept as large as practicable, as in the case of *Fig.* 2. Together with the difficulty of cold setting such springs, this affords some justification for the practice of using lower working stresses in tension springs as compared to similar compression springs.

Designing for Static Loads

Analysis Neglecting Curvature Effects: Where static loads or loads repeated but a few times during the spring life are involved, it is common practice in design to neglect curvature effects (which result in localized peak stresses at the inside of the coil) in calculating stress. If it is assumed that the effect of the direct-shear stress under yielding conditions (due to the load component *P cos* 60°, *Fig. 4b*) may be compensated for by a redistribution of stress, the stress τ_2 of Equation 8 is taken as zero. Similarly, if the direct-tension stress (due to the component *P sin* 60°, *Fig. 4a*) is neglected for static loading, the stress σ_t of Equation 4 is taken as zero. This amounts to taking the factors K, K_1, and K_2 in Equations 13 and 15 as unity. The result is the same as if c were taken as infinity, which would give a value of equivalent stress from Table I equal to about 15 per cent above the nominal value (i.e. $C_1 = 1.155$). If static loading is assumed and if curvature, direct shear and direct tension stresses are neglected this means that the 60 degrees end turn of *Fig.* 3 would be about 15 per cent weaker than the body of the spring. For lower index springs this weakening effect is thus somewhat higher than that to be expected if curvature is considered.

Considerations for Sharp Bends

End Loop Turned Up Sharply: For an end loop turned up sharply as in *Fig. 1b*, where the radius r_1 is very small, a rough analysis of stress may be made as follows (curvature effects included): The bending moment at A' (*Fig. 1a*) where the sharp bend begins is Pr approximately and the section modulus is $\pi d^3/32$. Assuming a factor of stress concentration K'_2 to take into account curvature at the radius r_2 (*Fig. 1a*) the bending stress at A' becomes

$$\sigma_b = K_2'\left(\frac{32Pr}{\pi d^3}\right) \qquad (16)$$

An approximate expression for K'_2 is (3)

$$K_2' = \frac{4c_2 - 1}{4c_2 - 4} \qquad (17)$$

where $c_2 = 2r_2/d$. (Usually the radius r_2 will differ but little from the coil radius r).

To this bending stress must be added the direct-tension stress $4P/\pi d^2$ due to the load P. Adding this to σ_b as given by Equation 16 the maximum stress σ at A' becomes

$$\sigma = K_2'\left(\frac{32Pr}{\pi d^3}\right) + \frac{4P}{\pi d^2} \qquad (18)$$

Taking the shear-energy theory as a basis, the equivalent shear stress τ_e at A' will be 57.7 per cent of the bending stress σ. Substituting this value in Equation 18 and simplifying, the equivalent stress at A' becomes

$$\tau_e = \tau_n\left[\frac{1.15\left(K_2' + \frac{1}{4c}\right)}{K}\right] \qquad (19)$$

where τ_n is given by Equation 10*a* and is the stress figured in the usual way including curvature. For large index springs where K'_2 and K approach unity and where $r_2 \approxeq r$ this equation indicates stresses around 10 to 15 per cent above the nominal stress τ_n in the coils.

If static loading is assumed and curvature, direct-tension and direct-shear effects are neglected, this is equivalent to taking $K = 1$, $K'_2 = 1$ and $c = \infty$ in Equation 19. This thus indicates an equivalent stress about 15 per cent higher than the nominal stress. In other words for the end loop of *Fig.* 1, a weakening effect may be expected even for static loading conditions where curvature is neglected. This conclusion may be modified, however, by consideration of the redistribution of stress when yielding of the material occurs.

Higher stresses may be expected at point A (*Fig. 1b*) where the spring joins the body of the coil. The stress at this point may be estimated as follows: Since the loop is assumed to be bent up sharply so that the mean radius r_1 at A is very small the stress at the inside of the bend at A will be principally torsion. In this case the stress concentration factor as taken from curved-bar theory will be given approximately by Equation 3 taking $c_1 = 2r_1/d$ instead of c. This gives for the peak shear stress at A

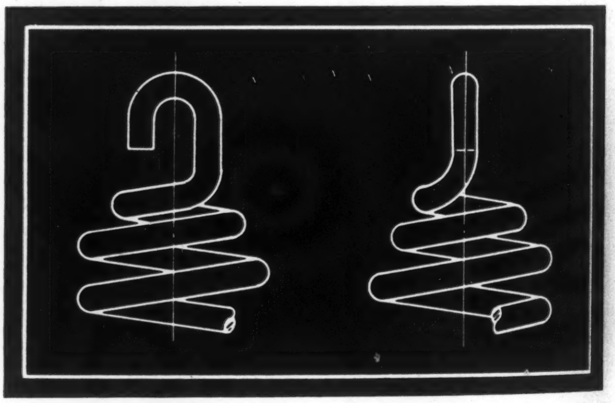

Fig. 5—End-coil design for reduced stress

$$\tau = K_1'\frac{16Pr}{\pi d^3} \qquad (20)$$

where K'_1 represents a stress concentration effect and is approximately, for $c_1 \geqq 3$,

$$K_1' = \frac{4c_1 - 1}{4c_1 - 4} \qquad (21)$$

If r_1 is small, the stress concentration factor K'_1 will be relatively large. (Thus $c_1 = 3$ and $K'_1 = 1.37$ for $r_1 = 1.5d$). This calculation shows the desirability of avoiding a small radius of curvature such as r_1 (*Fig.* 1), particularly where fatigue conditions are involved. This could be done by using a properly made full loop as shown in *Fig.* 2.

Between points A' and A (*Fig.* 1) there will be a combination of bending and torsion stresses which depend on the shape of the bend as well as on the radii r_1 and r_2. Since the peaks of these bending and shearing stresses do not occur at the same point, the combination of the two presents considerable complication and will not be discussed here. For practical purposes Equations 19 and 20 are probably sufficient.

An interesting expedient (6) to reduce stresses in the end coils is shown in *Fig.* 5. In this case the diameters of the end coils are gradually reduced before the loop is formed. Then when the loop is bent up the moment arm of the load at the point where the curvature is greatest will be reduced, giving lower stresses. Such a design, though more expensive than the conventional, may be worth while where high stresses are unavoidable.

Determining Active Turns

Effect of End Coils on Deflection: To find the total number of active turns in a tension spring the number of turns between points where the loop begins is first determined. To this is added the deflection due to the end coils. Tests made by Sayre (7) indicate that a half coil turned up to form a loop as indicated in *Fig.* 1 is equivalent to .1-full-coil as far as deflection is concerned. Thus if a spring has n' turns between points where the loops start, the total active turns would be $n' + .2$, the extra .2-turn being the equivalent of two loops. This conclusion may be shown analytically as follows: The half coil turned up to form a loop is equivalent to the quarter bend of *Fig.* 6. The deflection of this bend due to a load P is (8),

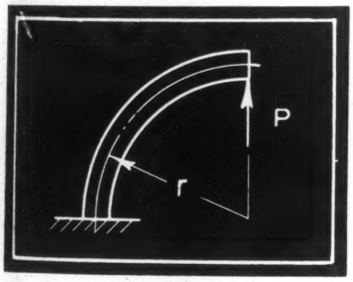

Fig. 6 — Quarter bend under load

$$\delta_1 = \frac{\pi}{4}\,\frac{Pr^3}{EI} \qquad (22)$$

where r = mean radius of loop (taken equal to coil radius), E = modulus of elasticity of material, I = moment of inertia of wire cross section. Taking $I = \pi d^4/64$ and the modulus of rigidity $G = E/2.6$ which is approximately true for most spring materials, by substitution in Equation 22, and simplifying the expression for δ_1 this becomes

$$\delta_1 = .096\left(\frac{64Pr^3}{Gd^4}\right) \qquad (23)$$

This is approximately .1 times the deflection per turn.

For the full coil turned up as shown in *Fig.* 2 the experimental work of Sayre (7) indicates a deflection equal to .5-turn. In this case the total number of active coils would be $n' + 1$ where n' is the number of coils between points where the loops begin.

Larger Indexes Reduce Initial Tension

Initial Tension: The amount of initial tension which can be put into a tension spring depends primarily on the spring index $2r/d$; the higher the index the lower the initial tension values. The values of stress corresponding to practical values of initial tension listed in Table II were published by Wallace Barnes Co. (9). These values are calculated from the usual spring formula with curvature effects being neglected. Hence the initial tension load P_1 may be figured from these values of stress using the formula

$$P_1 = \frac{\pi d^3 \tau_1}{16r} \qquad (24)$$

where τ_1 = initial tension stress.

As an example calculation, a tension spring has a 2-inch outside diameter and ¼-inch wire diameter so that the index $2r/d = 7$. When wound with maximum initial tension, from Table I the stress due to initial tension will be $\tau_1 = 16200$ lb/in.². From Equation 24 the initial tension load becomes

$$P_1 = \frac{\pi(16200)(.25)^3}{16(.87)} = 57 \text{ lb}$$

By changes in the method of winding, values of initial tension less than this may also be obtained.

It should be noted that no appreciable deflection of the spring will occur until the external load exceeds the initial tension load. After this the slope of the load-deflection characteristic will correspond to that of a similar spring without initial tension.

Table II

Torsional Stress Corresponding to Initial Tension

Spring Index ($2r/d$)	Initial Tension Stress (lb/in.²)
3	25,000
4	22.500
5	20,000
6	18,000
7	16.200
8	14.500
9	13,000
10	11,600
11	10 600
12	9,700
13	8.800
14	7,900
15	7,000

Reduces Working Stresses

Working Stresses: As indicated previously it is common practice to use working stresses for tension springs around 20 or 25 per cent below those used for similar compression springs. The analysis carried out previously indicates that for a loop turned up at 60 degrees as shown in *Fig.* 3, a weakening effect of 10 to 15 per cent may be expected as compared with a compression spring. As mentioned previously, this case is probably somewhat comparable to the full loop of *Fig.* 2. Because of this weakening effect and because of the difficulty of cold setting tension springs, a reduction of working stress by 20 or 25 per cent as compared with compression springs of similar wire size and material does not appear unreasonable. Further fatigue tests may modify this conclusion, however. Where the end loop is bent up sharply as indicated in *Fig.* 1, if fatigue conditions are present an even greater reduction of working stress may be advisable due to stress concentration at the sharp bend.

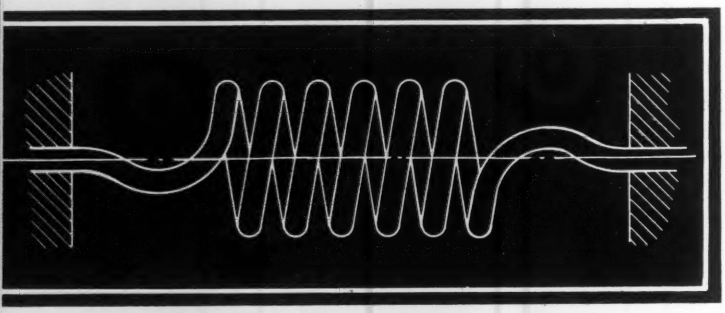

Fig. 7—Tension-compression spring with ends coinciding with spring axis

Tension-Compression Springs: An interesting example of a special spring which may carry both tension and compression loads is shown in *Fig.* 7. Such springs have proved useful in fatigue testing of machine parts where it is desired to apply alternating tension-compression loads. In forming such springs the wire or bar is brought out in a gradual loop so that the axis of the bar coincides with the spring axis as indicated. In this way stress-concentration effects due to sharp curvatures are avoided, as is important where repeated or fatigue loading is present.

A view showing the application of such springs in a turbine-blade fatigue-testing machine is shown in the head illustration. In this case nine springs, having the shape shown in *Fig.* 7, are clamped around the periphery of two circular plates. One of these plates is moved back and forth by means of a crank arrangement connected to the crosshead shown at the right of the testing machine. The advantage of this arrangement is that any distortion or plastic flow of the test specimen will not appreciably affect the spring loads.

In designing tension-compression springs of this type it should be borne in mind that if the stress is completely reversed, the stress range will be twice that of a spring subject to pulsating (zero to maximum) stress of the same peak value. Thus if the endurance limit is for example 70,000 pounds per square inch for springs tested in a zero to maximum range (this is an average value for springs of this type) in the case of a tension-compression spring the expected endurance limit would be about ±35,000 pounds per square inch for the same material. Additional stress concentration effects near the end turns would tend to reduce the endurance range below this value particularly if the curvature is sharp. On the other hand the endurance range in reversed stress may be somewhat greater than that to be expected for a zero to maximum type of stress application. In general, therefore, it appears that a working stress about half the allowable value for a zero to maximum range may be used for such springs.

In the machine shown the springs have an index of 3.5 and a wire size of ½-inch. Working stresses of ±25,000 pounds per square inch figured with curvature correction have been used without failure in service.

Consideration of the effects of stress redistribution during yielding may modify the conclusions reached in this analysis where static loading is involved.

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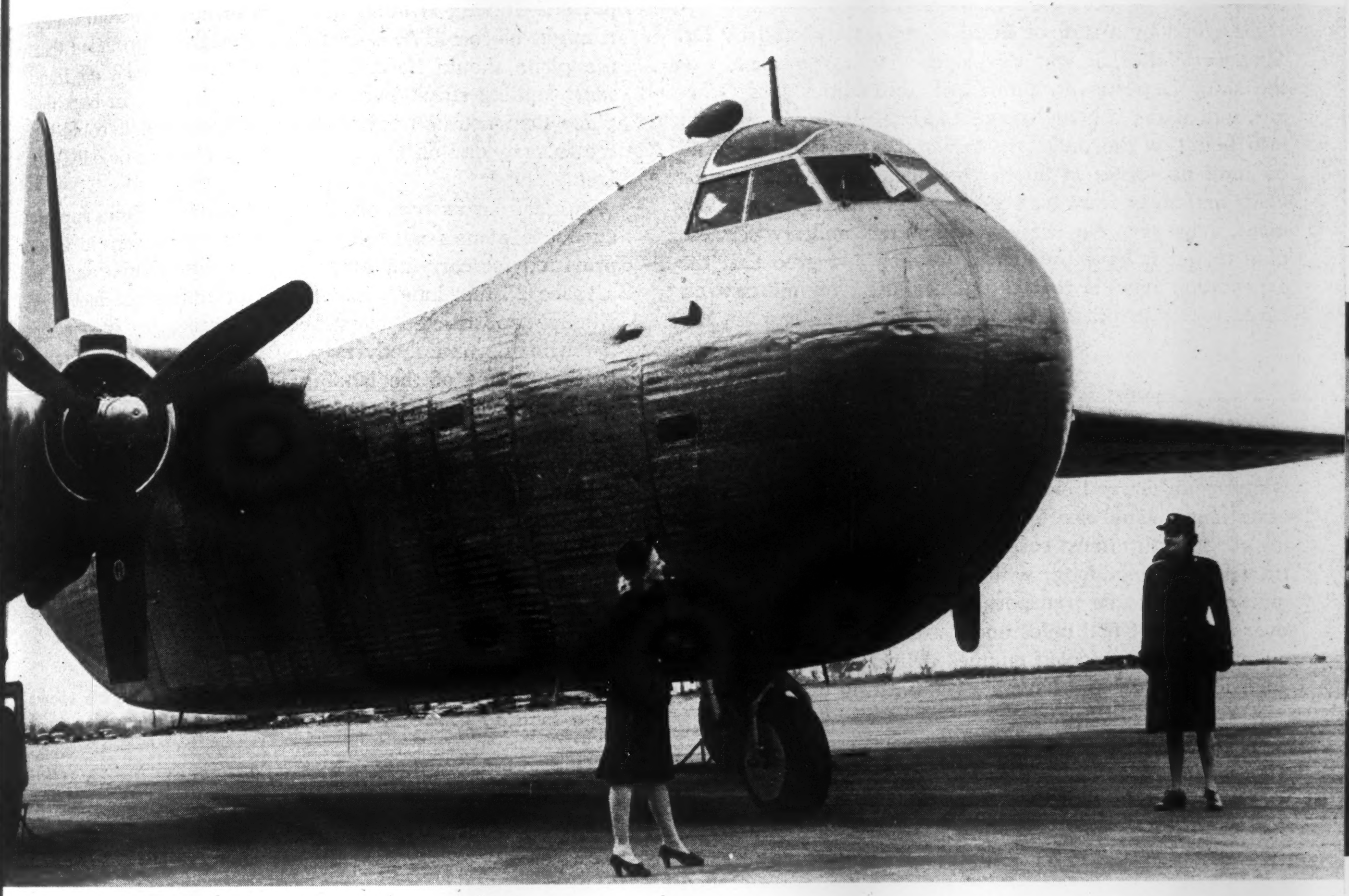
Armored Scout Car

COMBINING speed and maneuverability of an automobile with the punch and protection of a light tank, the autotank—identified by the Army as the M-8—is capable of battering down sizable trees and hurdling fallen trunks under perfect control. Designed by Ordnance and Ford, this armored scout car has seen action in Italy for some time although it was first demonstrated to the public recently in the dune country south of Chicago. Wheel suspension and drive are designed to provide engagement of all six wheels for maximum traction in the most difficult terrain. Transfer case and transmission which contribute materially to quick acceleration are manufactured by the Warner Gear division of Borg-Warner Corp.



Another Conestoga Makes Transportation History

A NEW air cargo plane, known officially as the "Conestoga" in tribute to the amazing vehicle which helped pioneer the settlement of colonial territories west of the Alleghenies, has been put into full scale production by the Edward G. Budd Manufacturing company in its new airplane plant near Bustleton on the outskirts of Philadelphia. Rated as one of the best-kept secrets of the war, this cargo carrier, shown in *Fig.* 1, is of a new type which utilizes welded stainless steel instead of riveted aluminum alloy sheets for the first time on any considerable scale. As such there is a strong possibility, in the view of aeronautical engineers, that the new plane will exercise a strong influence on the

Fig. 1—Above—New cargo-carrying plane, known as the Conestoga, employs welded stainless steel instead of riveted aluminum alloy sheet

character of air transports in the future.

Designed by a staff of Budd engineers directed by Dr. Michael Watter, it was developed under sponsorship of the Navy Department, Bureau of Aeronautics, and is being built under a Navy contract. The plane was brought into being at the outset to provide a type which could be used on routes in South America, where unimproved fields and short runways are common. Since its development, however, the uses to which the military services plan to put it have broadened to such a degree that the expectation now is that it will be used on military air cargo routes in practically all theaters of the war.

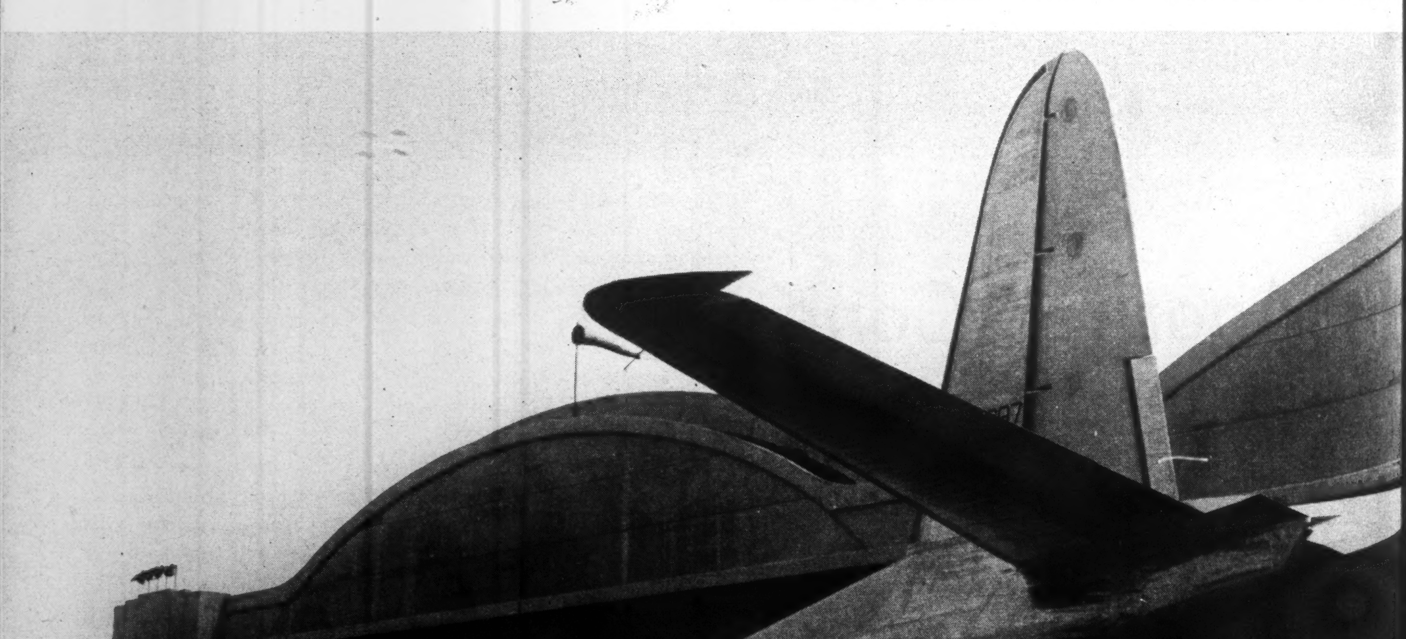
A Flying Box-Car

In creating the plane the engineers centered their efforts on the problem of designing a "flying box-car," which could carry freight economically on flights of moderate length, and also be capable of loading and unloading cargo with utmost ease and speed. It is a two-engine transport 68 feet long, with a wing spread of 100 feet, and is capable of transporting 10,400 pounds of cargo over a range of 650 miles under a C.A.A. license. Its 14-cylinder radial aircooled engines deliver 1200 horsepower each for take-off and have a normal rating of 1050 horsepower. Having in mind the types of airports from which it might be forced to operate, the designers provided that the plane should be able to get into and take off from short landing strips, even with heavy loads. In tests the plane demonstrated that in normal operation it could lift the full payload off the landing strip in a run of only 920 feet.

Aside from its type of construction and unique appearance the plane's most distinguishing characteristic is its provision for carrying cargo. The cargo compartment is a space 25 feet long, eight feet wide and eight feet high wholly unobstructed by structural members. The cargo hatch in the underbody may be lowered to form a ramp to the surface of the landing field, *Fig.* 2, or a loading platform level with the platform of a truck. Back of it are two large clearance doors which may be lifted out of the way to provide the necessary headroom to utilize the 8-foot-square cross section of the compartment in which the cargo is stowed.

The upward sweep of the afterbody of the plane, *Fig.* 3, provides a large clearance below the fuselage and back of the ramp to allow trucks to be moved close to the

Fig. 2—Hatch may be lowered to form a ramp as shown or a loading platform level with the platform of a truck



fuselage so as pick up or discharge shipments. Provision is made for handling cargo with the aid of a specially designed hoist fitted above the loading ramp, and a manually operated winch mounted on the forward end of the cargo compartment. These permit the loading of large, heavy pieces of freight, machinery or motor transport. Both an ambulance and a jeep can be rolled into the cargo space at once and still leave a large space for stowage of other cargo.

Michael Watter, Designer of the Conestoga

CHIEF ENGINEER of the Budd company's aircraft division, Dr. Watter, under whose direction the Conestoga was designed, obtained his engineering education in Russia and Italy. A native of the former country, he now is a citizen of the United States and a lieutenant commander in the U. S. Navy Reserve. The first planes Dr. Watter designed in this country were the original Corsair series built by the Chance Vought Corp. Of these, the 02U-1 recaptured three world records for the U. S. in 1927, while the X04-U-1 was one of the early all-metal airplanes designed and built in this country.

In 1933-34 he designed a long-range plane for the Mexican government. Later, as chief engineer for Burnelli Aircraft Ltd., he designed high-performance all-metal bombardment and commercial transport planes.

When with Glenn L. Martin company—1936-1939—Dr. Watter participated in and won two competitions on flying boats. As the outcome of the first the company was awarded a contract for 22 boats of the type now known as the Mariner (PBM). The second competition, involving four-engine long-range flying boats, resulted in what is now known as the Mars.

Floor Always Level

One of the most advanced developments in the design of the plane for cargo handling is that all loading is done on the level. On the ground and in the air the floor of the cargo compartment remains horizontal which obviates the necessity of hauling freight up an inclined floor before stowage. This is made possible by the use of a tricycle landing gear.

The unusually large cargo space is made possible through a unique design for the attachment of the wing to the fuselage. The wing structure does not pass through the fuselage in conventional fashion. Instead, through a novel arrangement of the fuselage, the wings are attached to five special side frames joined across the body by shallow, rigid transverse members, thus providing headroom of 8 feet—2½ feet more than would have been possible with conventional structure.

Adaptable for Troop Carrying

Although primarily a cargo carrier, the plane can be converted to troop transport or to aerial ambulance use. Provision is made for installation of 24 seats and 24 stretchers. There is a possibility that the plane may find an important use in the evacuation of wounded from battle areas to base hospitals behind the lines. In such use the level loading feature will prove most advantageous, as stretcher cases must be handled in horizontal position to insure their comfort.

Doors in opposite sides of the fuselage permit launching of paratroops from both sides simultaneously, provision also being made for the discharge of cargo, or paratroop supplies and equipment, by parachute through the opened clearance doors in the aft portion of the body.

The plane requires a crew of two, a pilot and a copilot, whose quarters are in a comfortable, full-height operating compartment in the nose of the plane, *Fig.* 1. As much engineering attention was given to the design of this compartment as to the facilities for handling cargo. Pilots who have flown the plane complimented the designers on this feature. Dual controls are installed, as well as remote-control radio and facilities for blind landings and take-offs.

According to the test pilots the plane is easy to fly, having exceptional stability and well-coordinated controls, thus fulfilling one of the principal objectives of its designers, namely, to provide an airplane of its weight and power which could be flown by pilots of limited experience.

The effective way in which stainless steel has been

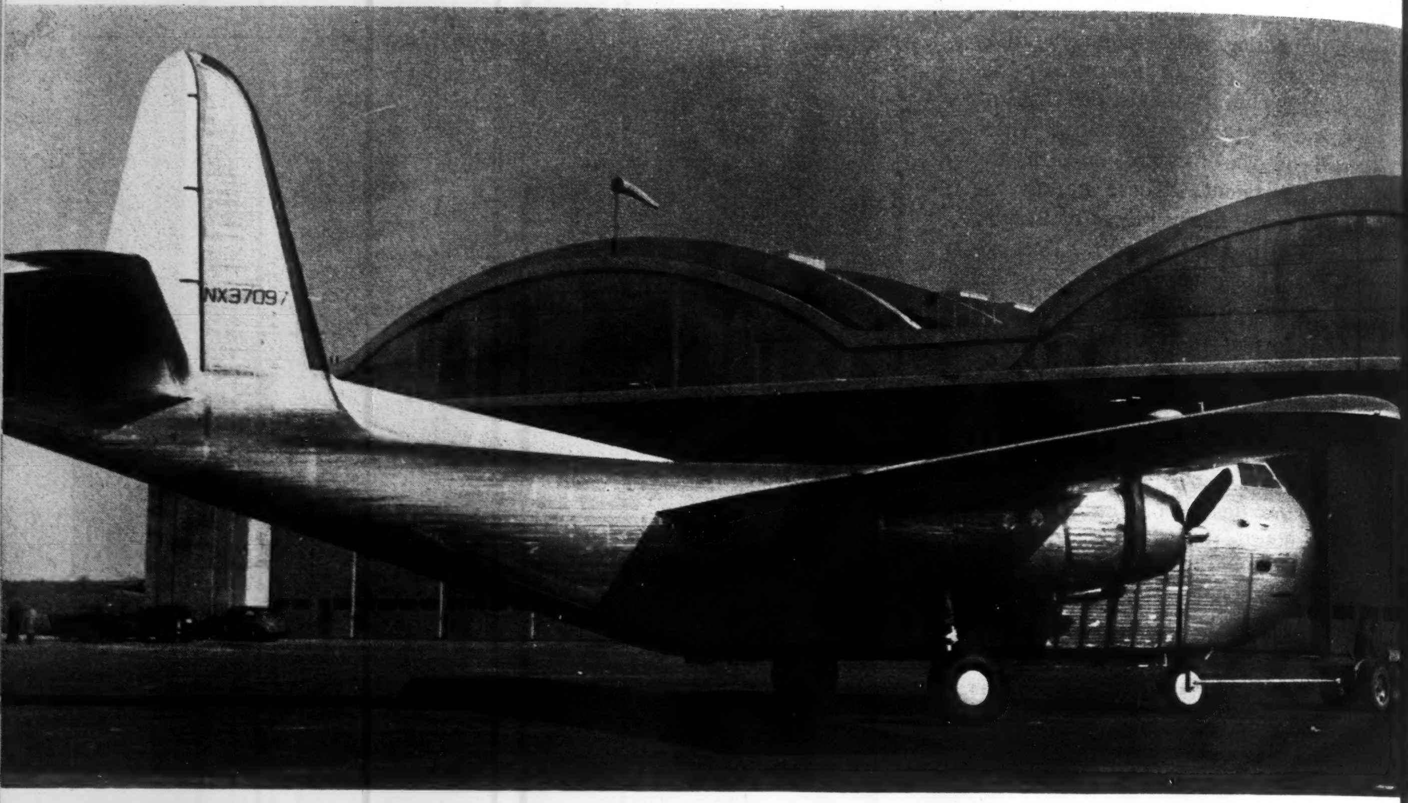


Fig. 3—Upward sweep of afterbody allows trucks to be moved close to the fuselage for loading and unloading

used is disclosed by the fact that, of the 33,800 pounds comprising the total weight of plane and full load, 10,400 pounds represents the weight of cargo. This is an unusually high ratio, 30.8 per cent, of total payload to total weight.

With a full load, including equipment, fuel, and the crew, the plane at a cruising speed of 165 miles per hour will have a range of better than 600 miles. The maximum range with available fuel tankage will be about 1700 miles. This can be increased up to 3500 miles with auxiliary fuel tanks. At cruising speed the two engines each will be required to deliver only 710 horsepower, thus providing an exceptionally economical consumption of fuel.

Stabilizing the Helicopter

INCORPORATING important principles which make for stability in flight and for precision control, a new helicopter has been developed by Bell Aircraft Corp. In this craft the rotor is rendered independent of the mast by the stabilizing device and the rotor tends to remain in a horizontal plane. This stability promises increased simplification of controls when such craft can be made available to the general public.

Air-Cooled Engine Drives Two-Bladed Rotor

Employing a two-bladed main rotor which makes possible a simplified hub design, the new helicopter is powered by a 160-horsepower Franklin engine of the horizontally opposed air-cooled type, mounted vertically immediately aft of the pilot's compartment. The closed-cabin model utilizes generous sections of transparent plastic, affording maximum vision. Rotor, mast transmission and engine are mounted integrally and are set in the fuselage in soft rubber mounts in order to reduce vibration.

Span of the rotor is 33 feet, the main rotor blades being of solid wood construction with a steel insert in the leading edge to aid in mass balancing and for added strength. The 5-foot diameter antitorque propeller is two-bladed, variable pitch and also has solid wood blades. It is driven by a tubular shaft and is controlled by a cable and pulleys.

Regarding the future of helicopters as private craft it is pointed out that although the controls are continually being revised and simplified, operation of such craft still requires a high degree of skill and technical knowledge. Some time will elapse before the operation of the helicopter can be simplified to a degree that the average automobile driver can operate one. At first helicopters will be operated as pleasure craft and by sportsmen. At some time later they will become generally available in the mass market, but only after considerable research and development. Inasmuch as the helicopter is still in its early stage of development, it is felt that the future of this type of craft can only be impaired if it is oversold and if too much is expected of it.

Developing the Hydraulic Torque Converter

By A. Lysholm
Stockholm, Sweden

WHILE the Lysholm-Smith torque converter of today employs the well-known Fottinger coaxial arrangement, it incorporates multiple-stage turbine blading which enables the torque-speed range to become automatically variable within a wide range of high efficiency. In order to present a clear picture of how this development was brought about and how it functions, the following discussion is devoted to the theory of the torque converter, design features of various types of hydraulic drives, and some details of early

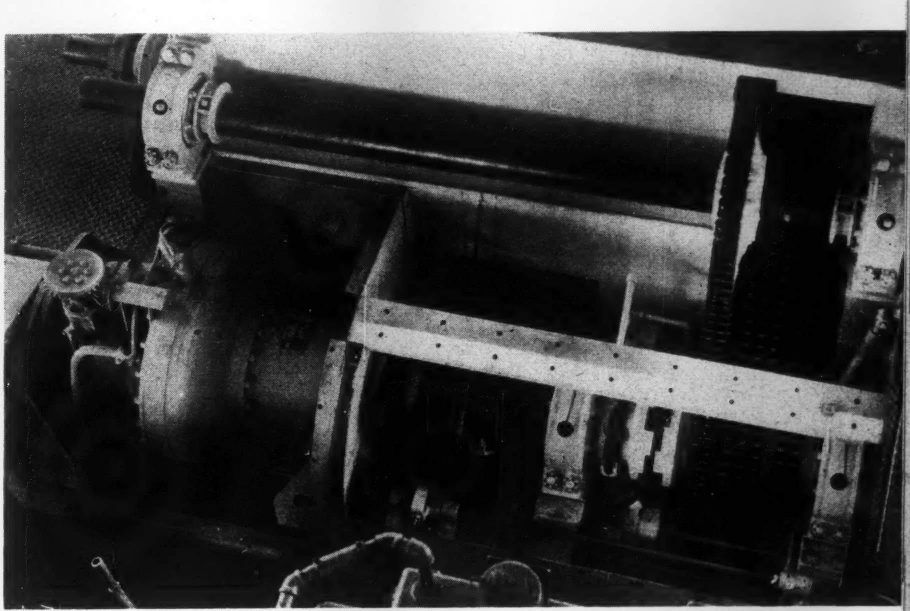
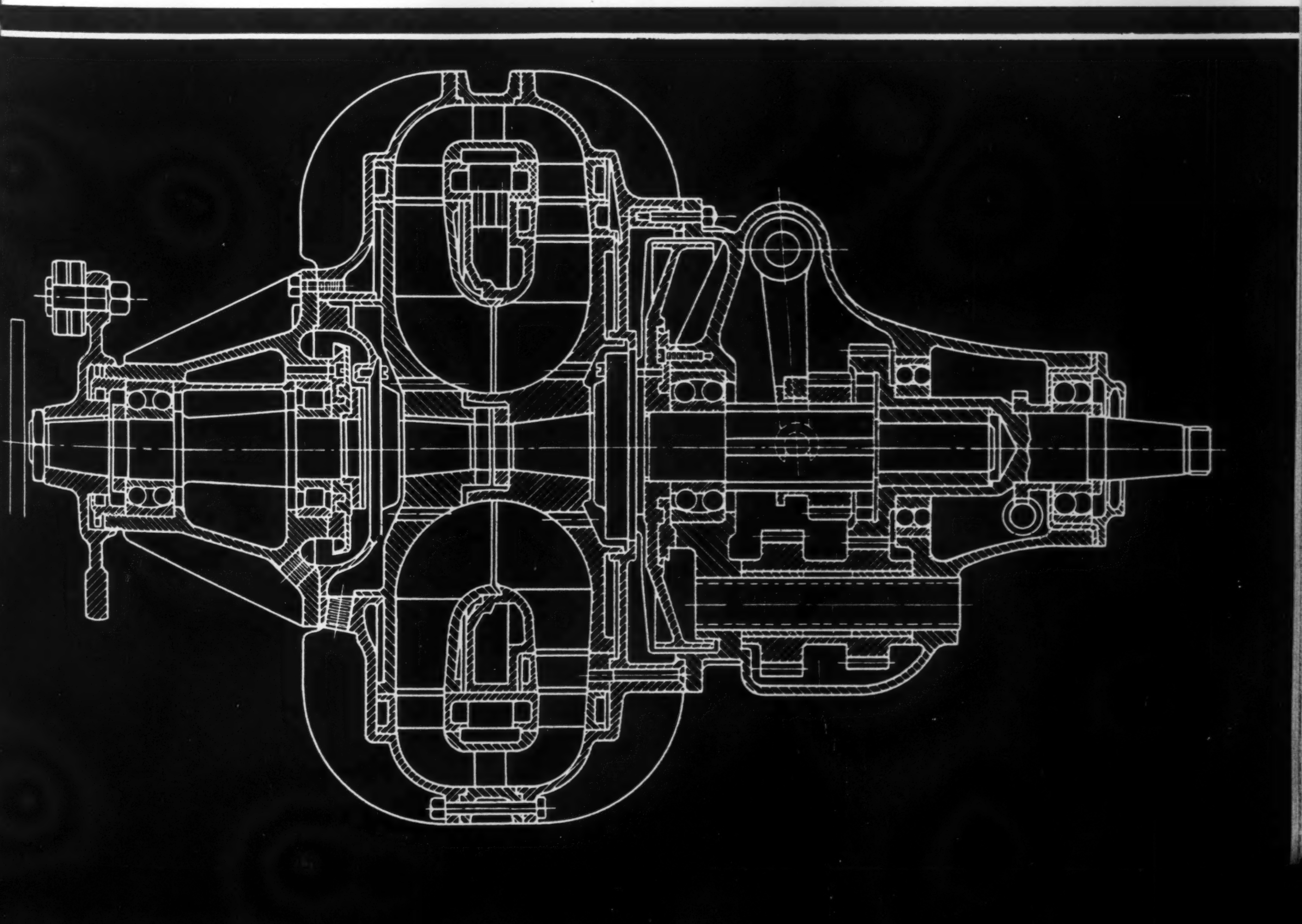


Fig. 1—Above—Installation of twin-disk adjustable-blade converter. Largest unit of this type to date was built for 1200-horsepower diesel locomotive

Fig. 2—Below—Original Lysholm-Smith test converter has one pump impeller, three rotating turbine rings and two stationary guide-blade rings



Abstract of a paper presented at the annual meeting of The American Society of Mechanical Engineers in New York.

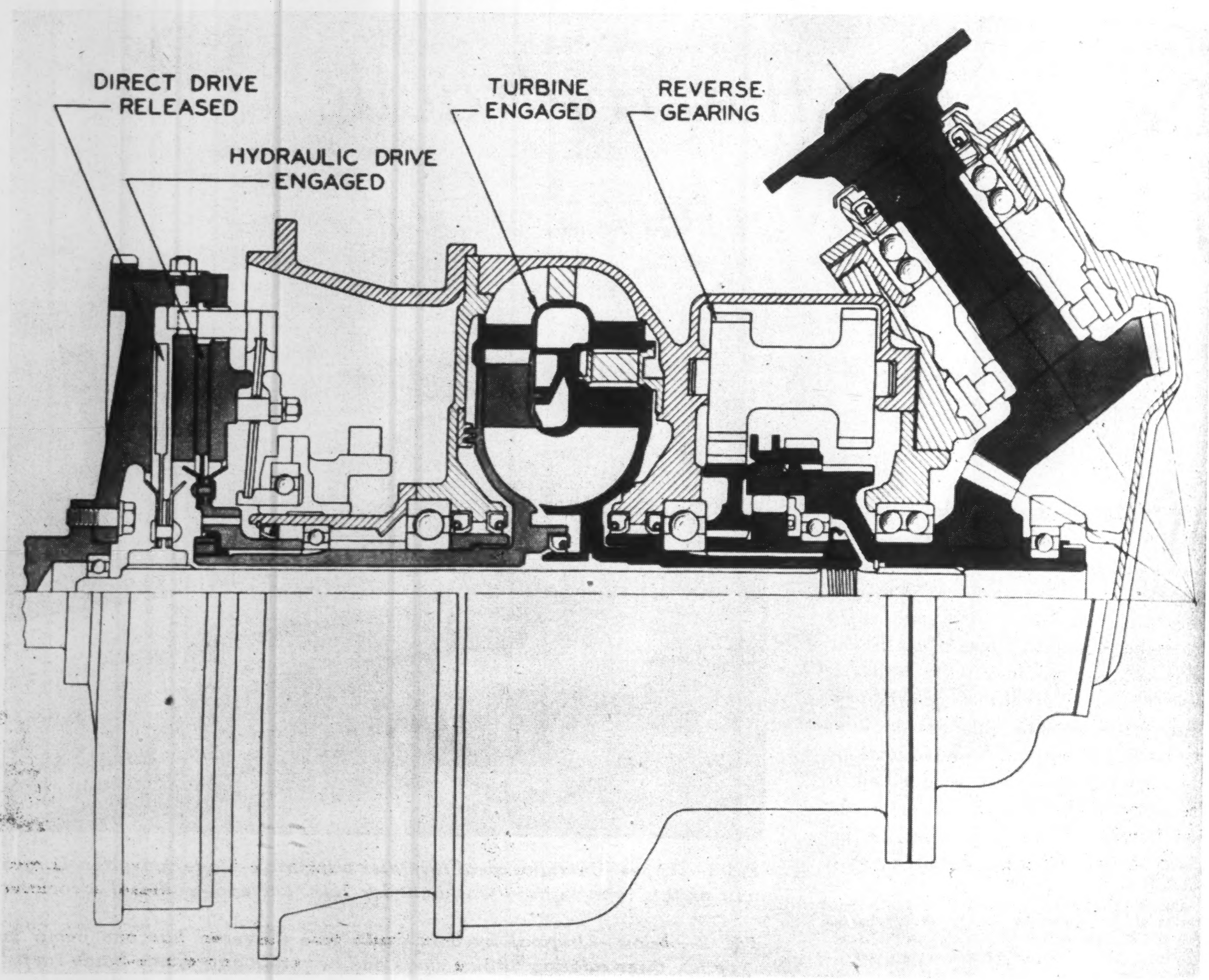

DIRECT DRIVE RELEASED
HYDRAULIC DRIVE ENGAGED
TURBINE ENGAGED
REVERSE GEARING

Fig. 3—Above—Direct-drive type of hydraulic transmission is shown fitted with bevel gearing for bus operation

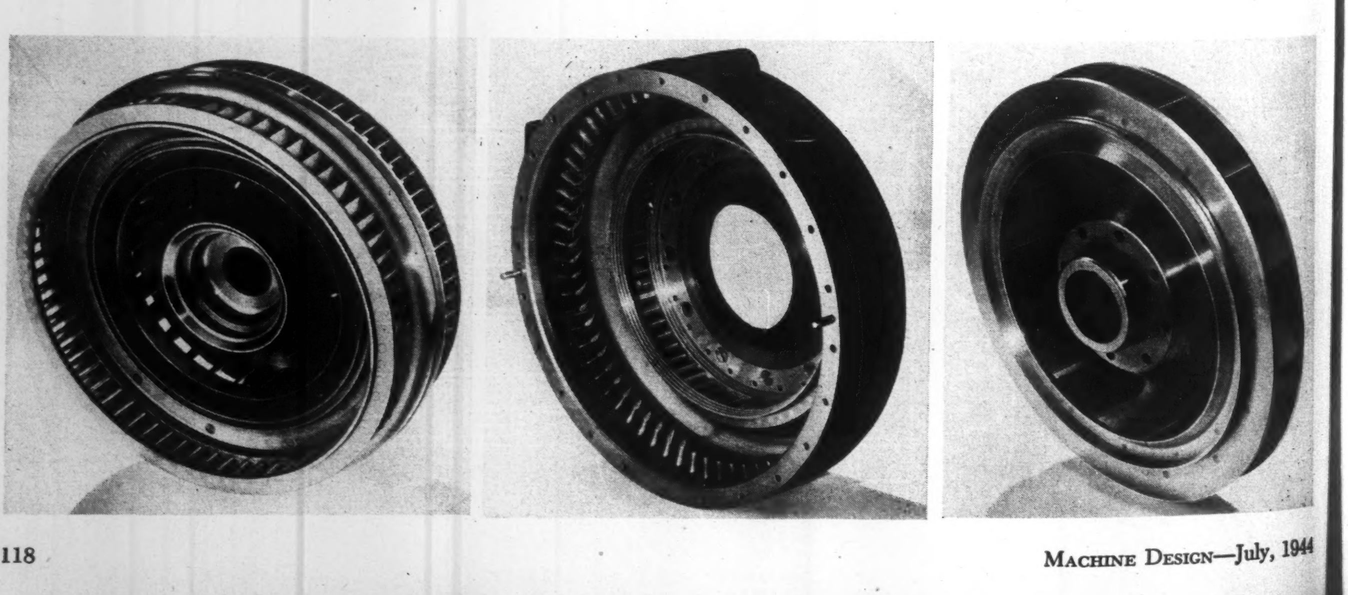
experiments in Sweden.

In the design of any efficient and mechanically satisfactory torque converter, the following considerations are important:

1. It is necessary to prevent the short-circuiting of the fluid between the revolving parts and the casing. This is done by a number of labyrinth seals as shown in *Figs.* 3 and 4.

2. Seals must be provided to prevent loss of fluid. These seals are normally of the axial-sealing type generally using a steel ring against a car-

Fig. 4—Below—Major component parts of modern converter. Labyrinth seals prevent short-circuiting of the fluid



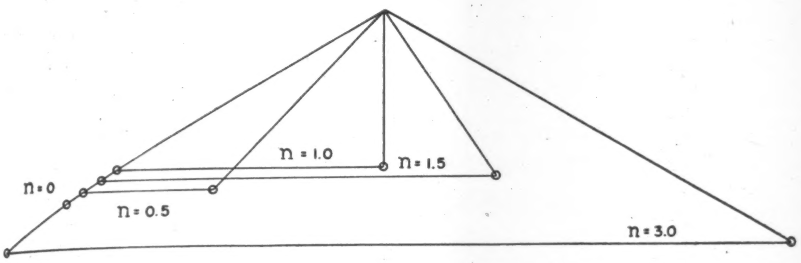
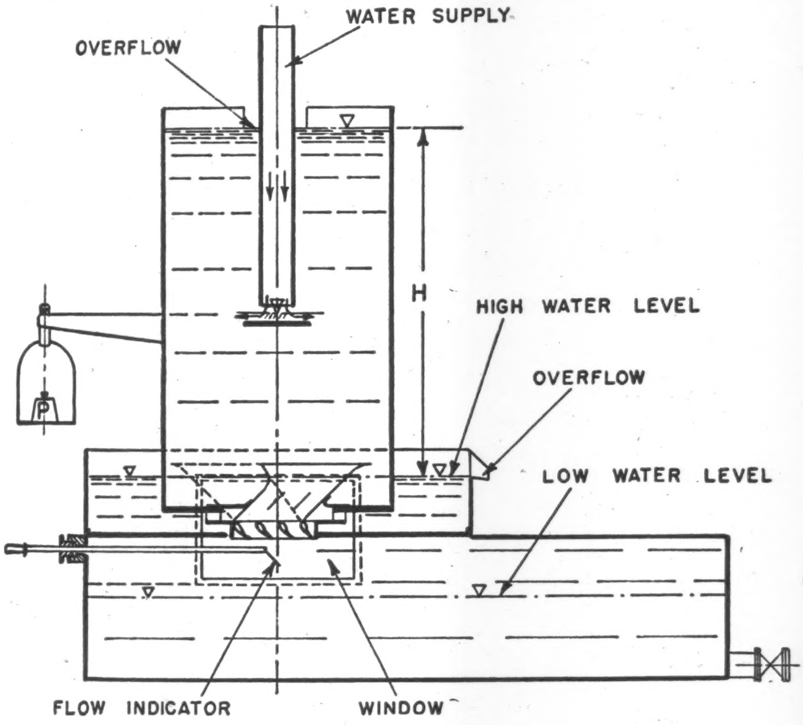


Fig. 5—Above—Velocity triangles with $Q\Sigma\Delta h$ constant. (n = relative speed.

Fig. 6—Below—Reaction balance employed to investigate blade friction at different speeds. By its use, friction factors at various inlet angles can be studied. Results of such tests are shown in curves of Fig. 7



bon ring for sealing the surfaces having difference in rotational speed. One of these rings has axial movement by means of a copper bellows or a synthetic-rubber diaphragm.

3. Evacuation of gas from the converter must be provided for. New fluid under certain conditions of circulation and temperature is liable to gasify to a slight degree. To insure the removal of this gas, it is drawn through a small pipe from the top of the converter.

4. It is very necessary, in order to prevent cavitation, that the converter be kept full of fluid under pressure. The automatic replacement of fresh fluid is insured by an injector or mechanically driven pump which draws it from a reserve tank as required and feeds it under pressure to the converter.

5. To provide for a temperature rise in the working fluid which may result from prolonged use of the converter, a small cooler is placed in the fluid circulation. This cooler circuit also includes a filter.

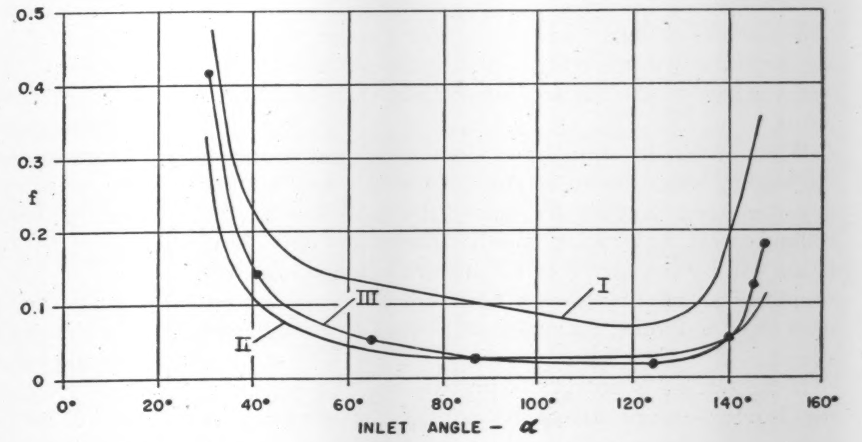


Fig. 7—Right—Test values for friction factors. I applies to profile No. 3. II and III apply to profile No. 2b

With the Lysholm-Smith torque converter which, like all modern hydrodynamic converters, incorporates the Fottinger coaxial arrangement, it has been possible to obtain high efficiency as well as a flat efficiency curve—principally by means of improved multiple-stage turbine blading—which enables the torque-speed range to become automatically variable within a wide range of high efficiency. *Fig.* 2 shows the original test converter which has one pump impeller, three rotating turbine rings and two stationary guide-blade rings. An account will be given of some of the fundamental principles of the torque converter and the experimental work leading to the evolution of the present types; namely, the fixed-blade type, the adjustable-blade type, and the type with direct drive.

Principal Losses Summarized

In a hydraulic gear of the turbine type, the following are the principal normal losses:

1. Friction loss in the pump wheel
2. Carryover loss between pump and turbine
3. Friction loss which occurs in the turbine blading
4. Carryover losses between the turbine stages
5. Carryover loss between turbine and pump
6. Leakage losses
7. Rotational losses
8. Mechanical losses
9. Auxiliaries.

A summary of these losses is given in the table "Balance of Losses" on Page 121.

Friction losses in the pump wheel and turbine in the

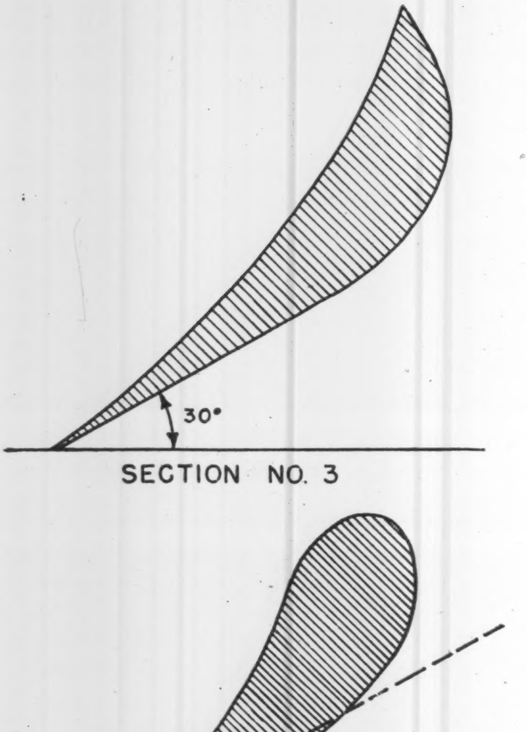


Fig. 8 — Left — Blade section No. 3 is reaction blade of ordinary design. Section 2b is especially suitable for varying inlet angles

Fig. 9—Below—Torque and efficiency curves of original converter at a constant input of 50 horsepower

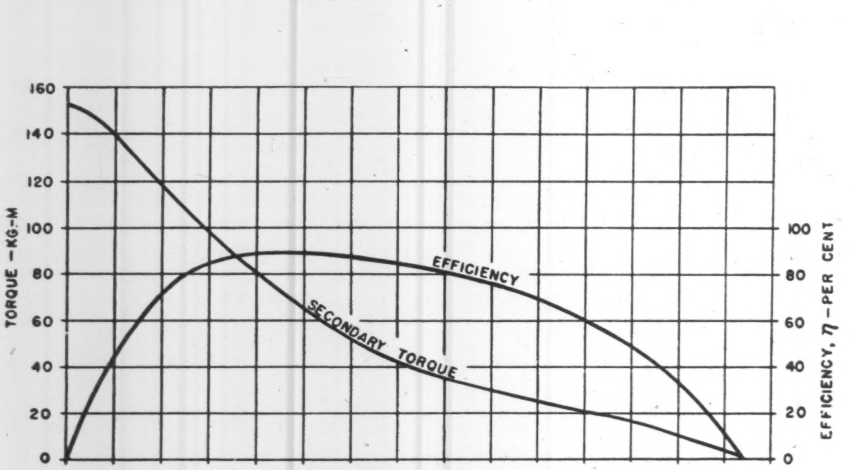


table include all losses due to the flow through the blading, together with the eddy-current losses at the inlet and outlet edges of the blades, but do not include leakage losses.

When running at different ratios of torque-speed, the blading losses due to varying inlet angles are the most important and account for nearly the total loss at the stalling point and racing speed. Therefore, only these losses will be considered in the following analysis, as this simplified assumption gives a fairly good view of the operation of a torque converter at a number of different speeds.

If the turbine is provided with only one runner, and if the leaving energy cannot be utilized, its efficiency at different speeds of the turbine shaft and with a constant supply of energy will, according to the principles of hydrodynamics, vary in accordance with a parabola. Conditions will be entirely different, however, if the leaving energy from each wheel is utilized fully in the next wheel. In the subsequent calculation, it is shown how the turbine "frictional" efficiency then will vary. Strictly speaking, the calculation is valid only for an axial-flow turbine and an infinite number of stages.

Calculating Pressure Drop

If c is the velocity of discharge from a blade and w designates the inlet velocity, then the useful pressure drop in the blade will be

$$\Delta h_u = \frac{c^2 - w^2}{2g} \quad \ldots\ldots\ldots\ldots (1)$$

The pressure drop on account of the friction can be expressed by the following formula:

$$\Delta h_f = \frac{f}{2} \times \frac{c^2 + w^2}{2g} \quad \ldots\ldots\ldots\ldots (2)$$

where f is the friction coefficient, which also includes eddy losses.

Thus, the total pressure drop in the blade is

$$\Delta h = \Delta h_u + \Delta h_f = \frac{c^2 - w^2}{2g} + \frac{c^2 + w^2}{2g} \times \frac{f}{2} \quad (3)$$

The efficiency of each blade evidently will be obtained from the relation between Δh_u and Δh, or

$$\eta = \frac{\Delta h_u}{\Delta h} \quad \ldots\ldots\ldots\ldots (4)$$

The values of Δh_u and Δh are obtained from the foregoing equations.

In the description to be given, the pump will be considered as an energy producer, and the efficiency of the turbine at different speeds will be examined, assuming a constant energy supply, that is, a constant product of pressure drop and quantity to be produced by the pump and absorbed by the turbine. Thus, the pressure drop in the turbine, multiplied by the circulating-fluid quantity Q, must be constant, or

$$Q\Delta h = \text{constant}$$

Assuming for the moment that Q is constant, then if a triangle for c, w, and u is drawn, and the peripheral speed u is assumed to vary with the constant flow, the value of w will be seen to decrease from c, when the turbine is stationary, to a minimum value when the direction of w is perpendicular to u (in the following called the optimum point or the optimum speed), and then to increase to c again at racing speed. From Equations 1, 2, and 3, it will be seen that the value of Δh_u will vary between zero and a maximum value, the value of Δh_f between $f/2 \times (c^2/g)$ and a minimum value, and the total pressure drop Δh between $f/2 \times (c^2/g)$ and a maximum value. Thus, at maximum torque at the stalling point, and zero torque at racing speed, friction absorbs all available energy.

Flow Must Be Increased

To get a constant product of $Q \times \Delta h$, however, the flow of liquid must be increased at all speeds in relation to the flow at the optimum point. At increased flow, the pressure drop varies with the second power of the flow, and thus the energy consumption must vary with the third power of the velocity of flow. The increased flow at a pressure drop Δh in relation to the flow at the optimum point thus will be obtained from the following relation:

$$\sqrt[3]{\frac{\Delta h_{max}}{\Delta h}}$$

where Δh_{max} denotes the pressure drop at the optimum point.

Hence it is clear that the velocity-ratio triangles must be enlarged in the same degree as shown in *Fig.* 5, where the triangles at starting and racing, as well as at a few intermediate speeds, are given. The triangles are drawn for different "relative speed," that is, the speed of the turbine in relation to the speed of the turbine at the optimum point.

It may be argued that the number of stages is of no importance for the efficiency of the converter, as the leaving loss always is utilized as inlet energy of the pump, but that is not correct since a certain percentage of the leaving velocity always is lost. Furthermore, it is possible with a multiple-stage converter to get a suitable stalling torque without using too large a pump diameter and too much backward slinging of the pump blades which results in efficiency loss over the whole speed range.

When laying out the blading of a converter, the usual pump and turbine equations should be used; thus

$$H_p = \frac{y_p}{g}(U_2 C_{2_u} - U_1 C_{1_u})$$

$$H_t = \frac{1}{y_t \times g} \sum (U_1 C_{1_u} - U_2 C_{2_u})$$

where

H_p = Pump head
H_t = Turbine head
y_p = Pump efficiency
y_t = Turbine efficiency
C_{1u} = Peripheral component of absolute inlet velocity
U_1 = Peripheral velocity of inlet
C_{2u} = Peripheral component of absolute outlet velocity
U_2 = Peripheral velocity of outlet.

Obviously the head of the pump and turbine must be

Balance of Losses

Item	First Test Converter	Standard Bus Converter	Standard Railcar Converter
1 Friction. pump wheel	3.5	4.5	4.5
3 Friction, turbine	2.5	3.0	3.0
6 Leakage	2.5	3.5	3.5
7 Rotation	1.5	1.5	1.5
8 Mechanical	.5	1.5	1.0
2, 4, 5, 9 and others	.5	1.0	1.0
Efficiency, per cent	89	85	85.5

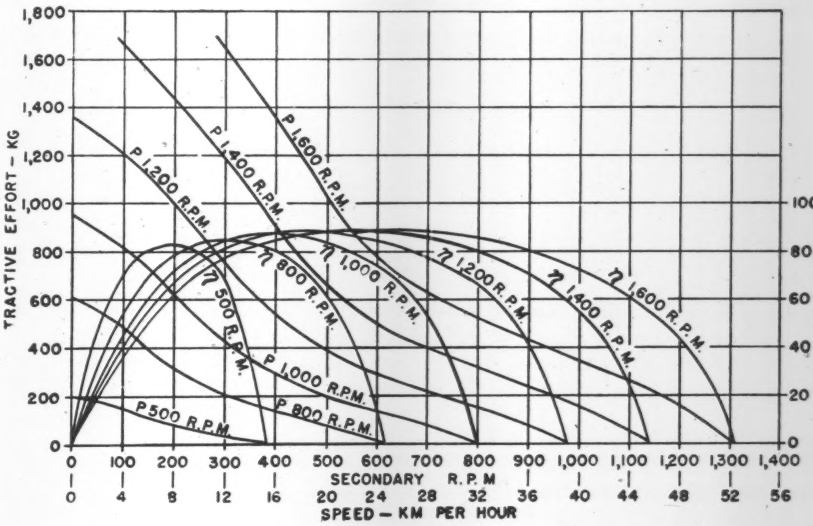
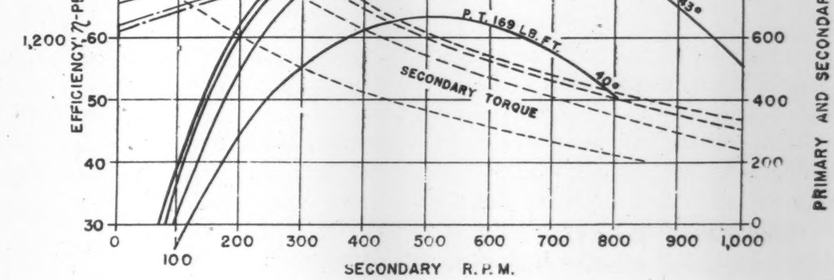


Fig. 10—Above—Tractive effort and efficiency curves of original converter apply to 50-hp omnibus; axle ratio, 1/4-1; effective wheel diam., 860 mm

Fig. 11—Below—Efficiency tests. Solid-line curves indicate converter fitted with 14-bladed pump wheel; dash-dot lines have blade angles given on curves; dotted-line curves are at constant primary torque



equal and, further, the exhaust energy of the pump as well as of the last turbine must be equal to the inlet energy of the turbine and pump, respectively, excepting of course, carryover losses.

In order to investigate the blade friction at different speeds, some preliminary tests were made with a reaction balance, *Fig.* 6, by means of which the friction factors at different inlet angles could be studied. The results for two blade-section investigations (Nos. 3 and 2*b*) are given in *Fig.* 7.

Blade section No. 3 in *Fig.* 8, is a reaction blade of ordinary design and section 2*b* is especially suitable for varying inlet angles. However, the last-mentioned type of blade proved to be even better at ideal inlet angles of flow. In *Fig.* 7 a curve is drawn, calculated from later tests made on the complete hydraulic converter with blades principally of section No. 2*b*.

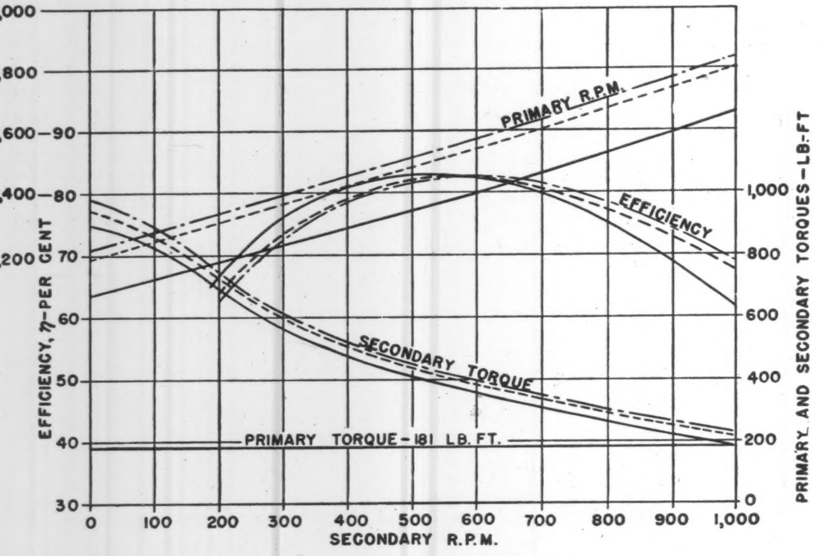
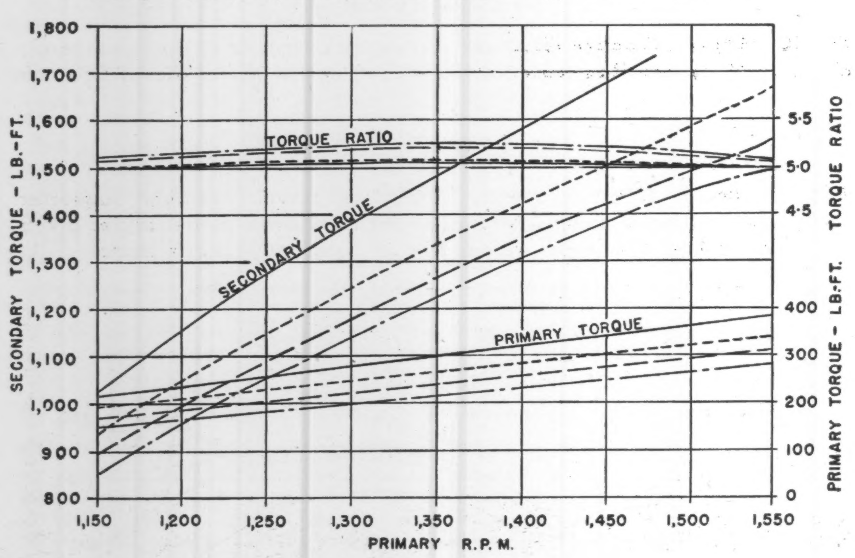


Fig. 12—Above—Speed-efficiency curves. Solid lines apply to standard pump wheel; dotted lines apply to standard pump wheel reduced 7/16-inch in diameter; dash-dot lines apply to standard pump wheel reduced 9/16-inch

Fig. 13—Below—Stalling tests on standard pump wheels, diameters reduced as follows: Solid lines, reduced 0; dotted lines, reduced 1/4-inch; dashed lines, reduced 7/16-inch; dash-dot lines, reduced 9/16-inch



As will be seen, the results obtained with the converter followed closely the test values with the reaction balance for inlet angles between 50 and 145 degrees. At angles smaller than 50 degrees, the real values of f are, however, higher than those obtained with the reaction balance, and it is thought that this is due to the fact that the number of blade channels in the reaction balance was too small. However, this test gave sufficiently good results to enable construction of the blade system of the complete gear. It is interesting to note that the blade friction closely corresponds to the friction loss in a smooth tube, as determined by "Blasius" tests.

Tests on Original Unit

The blade system in the original Swedish test gear, *Fig.* 2, consisted of a pump wheel and three rotating blade rings, as well as two fixed guide-blade rings in the stator. Performance tests were made by measuring input torque and stator reaction, as well as primary and secondary speeds.

Shown in *Fig.* 9 is the efficiency curve as a function of the turbine speed on the assumption that 50 horsepower is supplied on the primary side. This efficiency curve is flat and attains a maximum value of 89 per cent. The corresponding turbine-torque curve also is shown. This curve has a hyperbolic shape which differs considerably from the straight line corresponding to a parabolic efficiency curve.

During the tests, the velocity of the circulating flow of liquid also was measured with a Pitot tube at 800 and 1000 revolutions per minute of the primary shaft. From these measurements it appeared that, at a constant primary input, the flow of liquid at stalling speed was 22 per cent, and at racing, 50 per cent higher than at the speed corresponding to the optimum point.

Test Results Shown Differently

Test results in another form are shown in *Fig.* 10. Here the efficiency curves are drawn for various constant primary speeds from 500 to 1600 revolutions per minute as a function of the secondary speed. The maximum efficiency for 500 primary revolutions per minute is 83 per cent, and at 1400 revolutions per minute the efficiency has increased to a value of very nearly 89 per cent.

Converter Input Characteristics: A normal three-stage converter with short third turbine blades absorbs

(*Continued on Page* 157)

Applying Creep Data in Design

By Joseph Marin
Pennsylvania State College

Part I—Methods of Interpretation

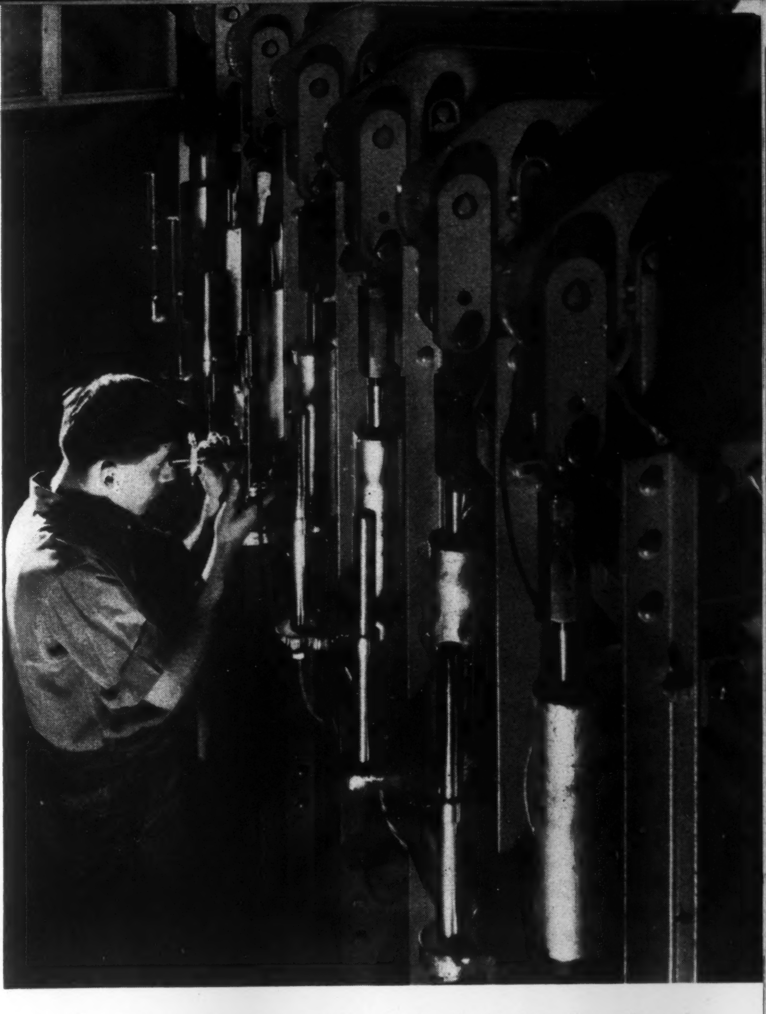
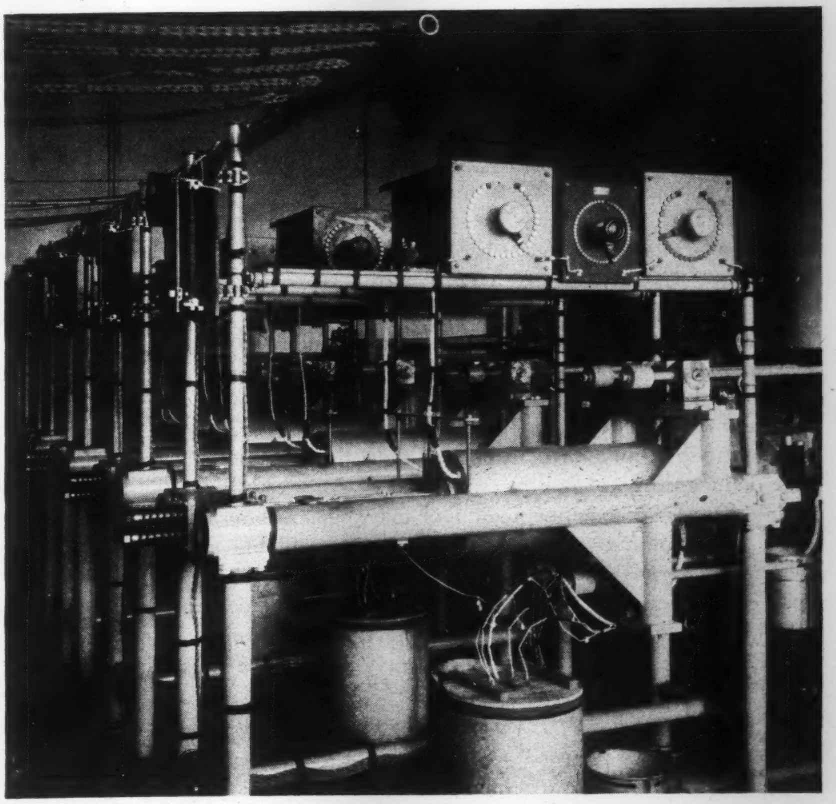


Fig. 1—Above—Tension creep testing machine employed by the Crane Co.

Fig. 2—Below—Testing machine used by General Electric for creep tests. Eleven-year tests have been run in this machine



SOME engineering materials at normal temperatures and others at elevated temperatures deform continuously with time when subjected to static or dynamic stresses, giving the condition called creep. The object of this article is to discuss the factors influencing creep, the methods of interpretation of creep tests, and the design of members subjected to creep.

A number of metals do not creep except at high stresses and high operating temperatures. Other metals such as lead and some aluminum alloys creep at low stresses and normal temperatures. The increase in the use of plastics necessitates a consideration of their creep properties since many plastics show creep at low stresses and normal temperatures.

An allowance for creep is particularly important in certain machine elements operating at high temperatures, such as turbine parts. For example, in turbine blades it is essential for proper operation to maintain small clearances between the moving and stationary parts. To provide for this clearance, information must be obtained on the creep deformations that reduce these clearances. Equipment employed in turbines and automotive and petroleum industries often is designed based on a working stress which considers permissible creep deformations.

In some constructions where plates

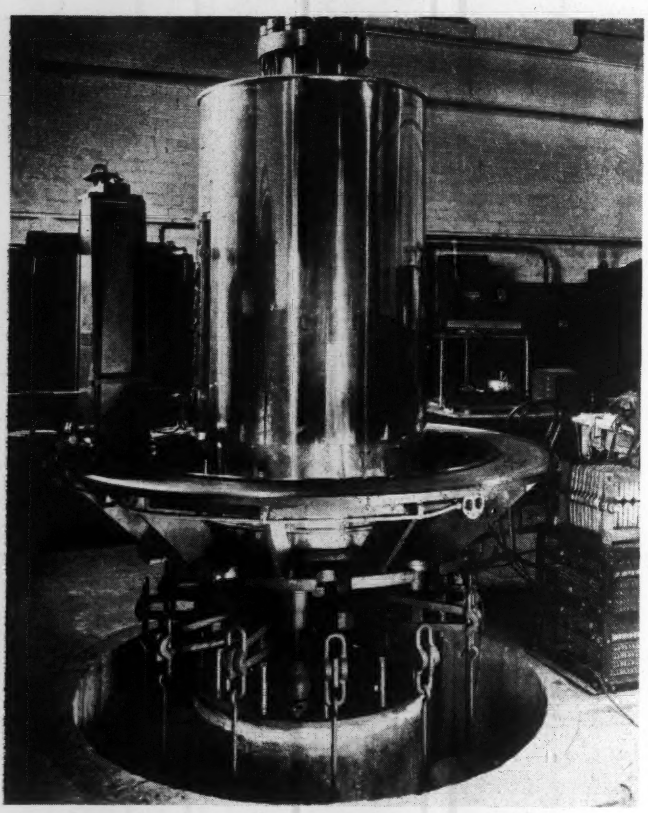


Fig. 3—Westinghouse cylindrical machine utilized for tension creep testing

are subject to compressive loads, creep may be an important factor in producing failure by buckling. On the other hand, some machine and structural parts may have large creep deformations without danger of failure as a result of lack of clearances or buckling. In these cases the working stress will be governed by the yield strength of the material for the particular temperature of operation. The effect of elevated temperatures on the yield strength of metals may sometimes be appreciable. The designer must therefore consider both the possible creep deformations and the yield strength in selecting the design stress to be used.

It is sometimes necessary to consider the possibility of rupture of the material due to excessive creep deformations. For example, if steam piping operating at a high temperature ruptures at an elongation of 25 per cent, a total increase in the pipe circumference of about 5 per cent could be allowed with safety. A total creep of 5 per cent in 200,000 hours (about 22.9 years) would give an increase of .5-inch in the diameter of a 10 inch pipe. This deformation would be too large and the actual allowable creep would have to be less than this, or on the order of magnitude of one or two per cent.

The stresses in machine members may be constant in value or fluctuating. Little information is available on creep in materials subjected to fluctuating stresses. In some cases the service condition in members subjected to static stresses is such that the stress does not remain constant but decreases in value with time. This stress condition is designated by the term *stress relaxation*. The bolts connecting the two flanges of piping operating at high temperatures is an important example of stress relaxation. The bolts are initially tightened so that a tensile stress is produced. As creep deformations in the bolts and flanges occur, the stress is reduced until leakage at the flanges is produced if the initial stress was not correctly selected.

Tests Simulate Service

Generally creep tests attempt to reproduce service conditions and to consider the effect of the variables stress, temperature and time on the creep deformation. Creep tests are also conducted for the purpose of selecting the most suitable alloy when several materials are available that satisfy the other design requirements such as corrosion resistance and fatigue strength.

The kind of stress produced is another important consideration in design of machine parts where creep is produced. Simple tensile, bending or torsional stresses may not always be produced and the condition of a combined state of stress often occurs. Most design stress values, however, are based on the simple tension test. For this reason the following discussion will be confined mainly to this type of stress.

In addition to the tension creep test it is usually necessary to determine the influence of the operating temperature on the yield strength of the material since the design stress selected must be based on both the strength of the material and the permissible creep deformation. Testing techniques to determine the effect of temperature on the mechanical properties in tension are described by Tapsell.[1] The testing procedure is tentatively outlined by the American Society for Testing Materials.[2] A good deal of the early test data giving values of the various mechanical properties as influenced by elevated temperatures is given in a "Symposium on the Effect of Temperature on the Properties of Metals" published by the A.S.M.E. and A.S.T.M. in June, 1931.

Constant Conditions Are Maintained

In the usual tension creep test a tension specimen is subjected to a constant stress and temperature. Equipment is provided for determining the creep deformation over a finite gage length at various intervals of time covering the entire duration of the test. The load is usually applied to the specimen by means of a lever system as shown in *Figs.* 1, 2 and 3. The elevated temperature—if this variable is considered—is provided generally by means of thermocouple heating units.

Most creep tests are conducted for a period of 1000 hours. Tests are sometimes conducted for 200 hours and a few have been reported which extended for a period of time exceeding 10 years. The test procedure for tension creep testing has been standardized to some extent by the A.S.T.M.[3] In order to determine the influence of the stress on the creep deformations several test units with different loads applied are used as shown in *Fig.* 1. In creep testing, measurements of load, temperature, creep deformation and time are recorded. From these measurements the unit creep deformation e can be plotted as a function of time t for a particular stress S and temperature T. *Fig.* 4 shows such plots for a medium carbon steel.

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[1] Tapsell, H. J.—*Creep of Metals,* Oxford Univ. Press, New York, 1931.

[2] "Short-Time Elevated-Temperature Tension Tests of Metallic Materials", Designation E21-42T, A.S.T.M. Tentative Standard, 1942, Page 1577.

[3] "Recommended Practice for Conducting Long-Time High-Temperature Tension Tests of Metallic Materials," A.S.T.M. Designation E22-41, 1942, Page 930.

The shape of these creep-time curves depends upon the material, the temperature and the stress considered. For metals at high values of stress and temperature four stages on the creep-time curve can be distinguished:

1. An initial stage in which the total deformation is partly elastic and partly plastic
2. A region where the rate of creep decreases with time due to strain hardening
3. A region in which the effect of strain hardening is counteracted by an annealing influence resulting in a constant or minimum creep rate
4. A final stage in which the creep rate (or slope of curve) increases until fracture occurs.

For the lower stresses and temperatures the fourth stage is not present and the creep-time relation is linear, giving a constant minimum creep rate or slope as shown in *Fig.* 4. The influence of such factors as crystal grain size, age-hardening, oxidization and corrosion have been discussed by various investigators.[4]

With the creep-time data obtained as in *Fig.* 4 there remains the problem of extrapolation of the test data to include periods of time not covered by the tests. This is the situation since the time covered by the test, which is usually about 1000 hours, is considerably less than the estimated service life of the part which is being designed. Some of the methods which have been proposed for the extrapolation of creep-test data will be discussed briefly.

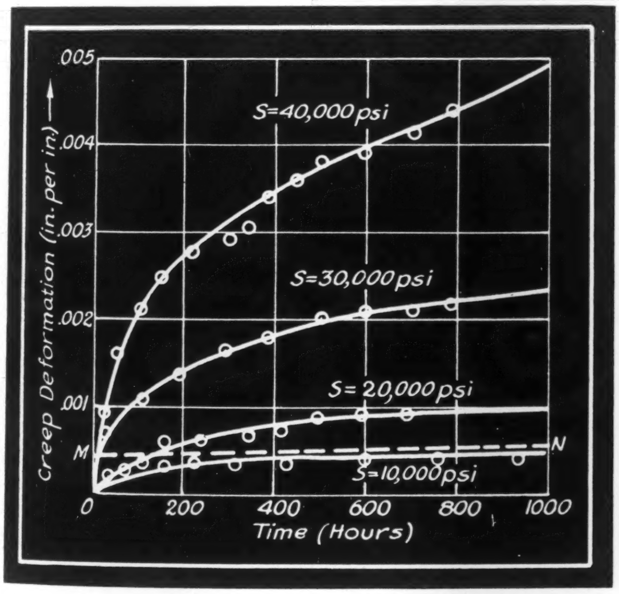

Fig. 4—Creep-time curves for 12 per cent chrome iron at 800 degrees Fahr. as tested by J. J. Kanter, Crane Co.

LOG-LOG METHOD: For the lower temperature and stress values the log-log method of interpretation has been most widely used. The creep-time relations of *Fig.* 4 can often be represented approximately by straight lines such as MN. Then the unit creep deformation e at any time t is expressed by

$$e=e_o+Ct \quad \text{.......} \quad (1)$$

where e is the intercept OM of the line MN on the creep axis, and $C=de/dt$ is the minimum constant creep rate or the slope of the line MN. Most of the methods used for the interpretation of creep test data assume the linear creep-time relationship expressed by Equation 1. It is also assumed that the relationship in Equation 1 can be extended to cover times beyond those covered by the tests.

By the log-log method the creep rates are calculated for each stress value and the relation between the creep rates and stresses plotted on a log-log plot as shown in *Fig.* 5 It is then assumed that the plotted points fall on a straight line. There is considerable evidence to show that this is a good assumption for the lower stress and temperature values.[5]

The straight line relations of *Fig.* 3 can be expressed by the equation

$$\log C=\log B+n \log S \quad \text{.......} \quad (2)$$

where B and n are experimental constants.

The usual and more convenient form for expressing Equation 2 is obtained by writing the antilog of Equation 2, namely,

$$C=BS^n \quad \text{.......} \quad (3)$$

Knowing the experimental constants B and n for a particular operating temperature T, Equation 3 can be used for selecting the design stress S. This can be done when the allowable creep rate C is selected. The selection of the creep rate as a basis for the design stress does not take into account the creep deformation represented by the intercept of the line MN on the creep axis, *Fig.* 5. This initial creep, however, may often be neglected since it is usually small compared to the final creep.

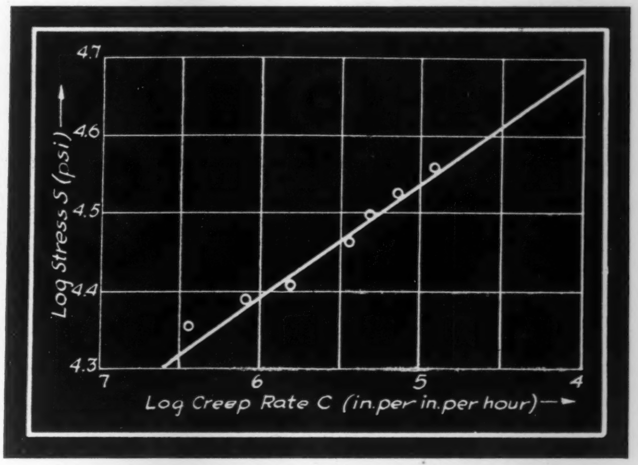

Fig. 5—Log-log-creep rate VS. stress relation for .3 per cent carbon cast steel, reported by R. W. Bailey, National Physical Laboratories, England

For comparison with other methods of interpretation, it is desirable to express Equation 2 in terms of the creep deformation and time. Placing $C=e/t$ in Equation 2,

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[4] Kanter, J. J.—*A.S.M. Transactions*, Vol. 24, No. 4, Dec. 1936, Page 870.

[5] "Compilation of Available High-Temperature Creep Characteristics of Metals and Alloys", *A.S.T.M.* and *A.S.M.E.*, March, 1938.

$$\log e=\log t+\log B+n\log S \tag{4}$$

To indicate the range of variation in the values of the experimental constants B and n, TABLE I gives values obtained for various steels tested at different temperatures. The tabulated values show a considerable spread in the possible magnitudes of both B and n.

TABLE I

Values of Experimental Constants

Material	Temperature (°F)	Reported by	B (in.^{2}/lb)^n/day	n
.39 C forged steel	752	McVetty	16×10^{-46}	8.6
.30 C steel	752	McVetty	48×10^{-38}	6.9
2.0 Ni, .8 Cr, .4 Mo steel	842	Weaver	20×10^{-23}	3.2
2.0 Ni, .8 Cr, .4 Mo steel	857	Weaver	10×10^{-20}	3.0
1.4 Mn, .3 C, cast steel	842	Weaver	79×10^{-27}	4.7
12.0 Cr, 3.0 W, .4 Mn, stainless iron	1022	Weaver	15×10^{-16}	1.9
Ni-Cr-Mo steel	932	Westinghouse Co.	91×10^{-20}	2.7
Ni-Cr-Mo steel	932	Westinghouse Co.	49×10^{-14}	1.3
12.0 Cr steel	850	McVetty	10×10^{-27}	4.4

LOG METHOD: The log method of interpretation is perhaps the one most commonly used for the higher stresses and temperatures. It is similar to the log-log method with the exception that the stress rather than the logarithm of the stress is plotted against the log of the creep rate C. The equation defining this relationship between creep rate and stress is:

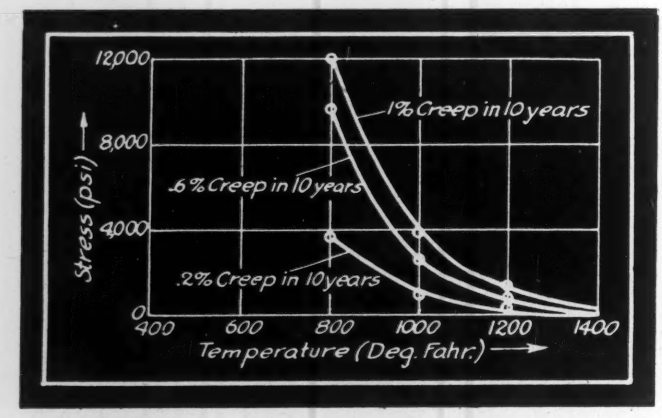

Fig. 6—Design stress-temperature relation for 12 per cent chrome steel according to Kanter

$$\log C=B_1+D_1S \tag{5}$$

where B_1 and D_1 are experimental constants.

In terms of the creep deformation $e=Ct$, Equation 5 becomes

$$\log e=B_1+D_1S+\log t \tag{6}$$

Refinements in defining the creep-time relationship have been proposed by several investigators. In place of the straight-line relation of Equation 1, McVetty[6] proposed the following equation as a better fit to the creep-time curves:

$$e=e_o+Ct-\frac{c}{q}\epsilon^{-qt} \tag{7}$$

where e_o is the creep intercept as in Equation 1, c and q are experimental constants, and ϵ is the base for Napierian logarithms.

In attempting to consider the effect of strain hardening on creep, Weaver[7] assumed that the creep rate varied inversely with the time, approaching a constant creep rate C. The empirical equation defining this condition was assumed as

$$e=a\log t-b+Ct \tag{8}$$

where a and b are experimental constants for a particular stress and temperature value.

Sturm[8] expressed a linear relation between the creep and time on a logarithmic plot. This result was based on a number of tests at room temperatures made on aluminum alloys, copper and steel wire. Sturm expresses the creep deformation by the equation

$$\log e=\log e'+k\log t \tag{9}$$

where e' and k are experimental constants for a particular creep-time curve.

The use of Equations 7, 8 or 9 in the determination of the design stress is complicated by the fact that a creep-time curve must be available for each stress. Two relations which include the variables stress and creep, as in the log-log and log methods, are those proposed by Soderberg[9] and Nadai.[10]

SODERBERG'S METHOD: In Soderberg's method it is assumed that for creep tests using various stress values and a constant temperature the creep-time relations are geometrically similar. Mathematically this means that the creep deformation e can be written

$$e=S'T' \tag{10}$$

where S' is a function of the stress only and T' is a function of the time only. In defining values of S' and T' it is assumed, furthermore, that the creep curves approach asymptotically a minimum creep rate and that arithmetical increments of stress produce geometrical increments in the creep rate.

These assumptions can be expressed by the equations

$$S'=A_1(\epsilon^{S/s_1}-1) \tag{11}$$

$$T'=t+T_0(1-\epsilon^{-t/T_1}) \tag{12}$$

where A_1, s_1, T_o and T_1 are experimental constants and ϵ is the base for Napierian logarithms. Substituting the values of S' and T' from Equations 11 and 12 in Equation 10, the creep deformation in terms of the time and

(*Concluded on Page* 182)

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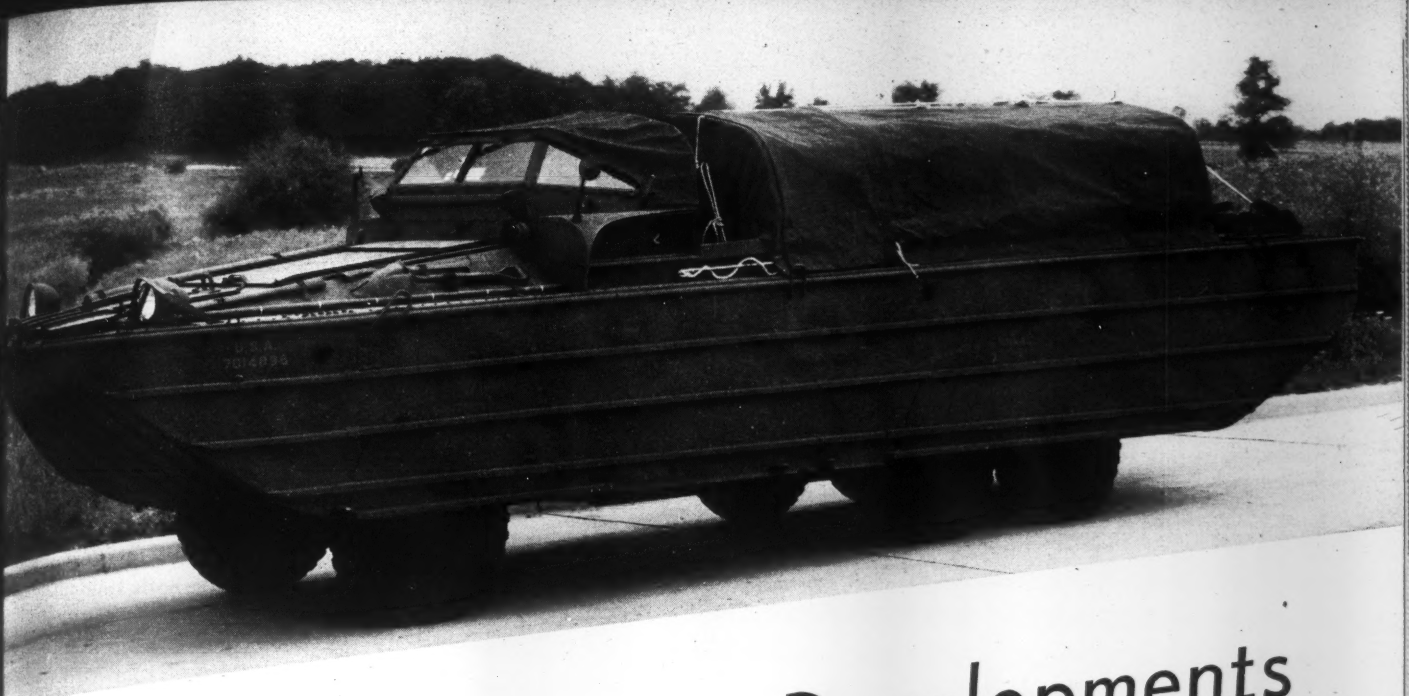
[6] McVetty, P. G.—"Working Stresses for High Temperature Service", *Mechanical Engineering*, Vol. 56, No. 3, March, 1934, Page 149.

[7] Weaver, S. H.—"The Creep Curve and Stability of Steels at Constant Stress and Temperature", *Transactions A.S.M.E.*, Vol. 58, No. 8, 1936, Page 745.

[8] Sturm, R. G., C. Dumont and F. M. Howell—"A Method of Analyzing Creep Data", *Transactions A.S.M.E.*, Vol. 58, 1936, Pages 62-67.

[9] Soderberg, C. R.—"The Interpretation of Creep Tests for Machine Design", *Transactions A.S.M.E.*, Vol. 58, No. 8, Page 733, 1936.

[10] Nadai, A.—"The Influence of Time Upon Creep—The Hyperbolic Sine Creep Law", *S. Timoshenko, 60th Anniv. Volume*, The MacMillan Company, New York, 1938, Page 155.



Effects of Wartime Developments On Future Steels

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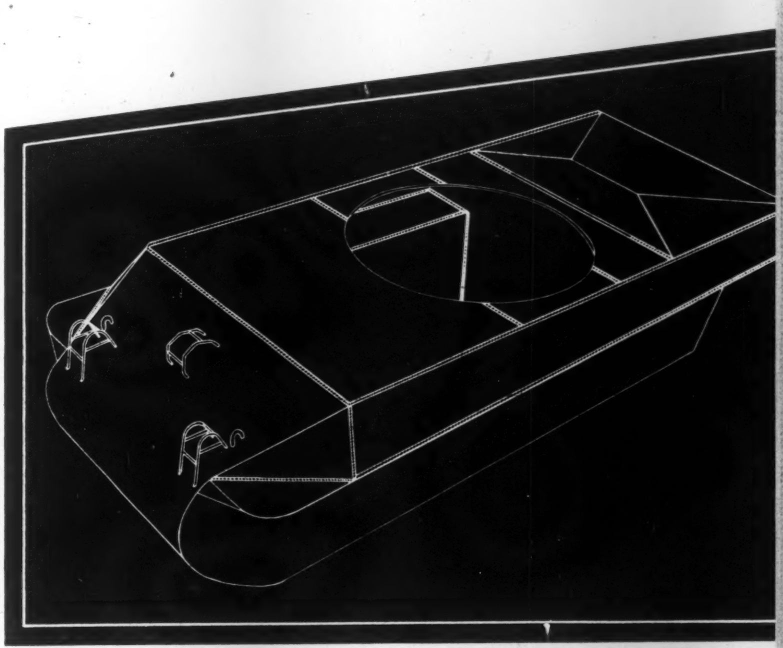

Fig. 1—Top—Entire hull of the two and one half-ton amphibian truck ("Duck") is a spot and arc-welded assembly of sheet steel

Fig. 2—Above—Sketch shows the hull structure of a military combat vehicle, an outstanding example of all-welded armor plate

THE WAR has intensified and accelerated certain types of scientific work, particularly those which help to translate fundamental knowledge into usable tools for engineers. Steels are participating in this intensification, and engineering practices are being improved thereby. An important aid in these accelerated studies has been the ignoring of normal peacetime competitive barriers, with resultant free interchange of information.

Some of the more important wartime developments pertaining to steels have been made along the following lines: Fatigue endurance, heat treatment, castings, welding, alloy evaluation, hardenability, special addition agents, and N.E. steels. Most of these developments were under way before the war began, but wartime progress in all these phases of steel metallurgy has been so rapid as to justify classing them as wartime developments.

The foregoing developments will be briefly summarized and discussed, primarily from the standpoint of the engineer. Predictions (guesses) will be made as to how our future practices with respect to steels will be affected. It must be remembered, however, that the war is far from finished, that developments discussed are not yet complete, and that therefore the influence of each development on postwar steels will depend upon (a) future progress made during the war, (b) status of each development during the period in which we return to what

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Abstract of a paper presented at the War Engineering annual meeting of the Society of Automotive Engineers in Detroit.

Table I

Suggested Mechanical Property Specification Ranges

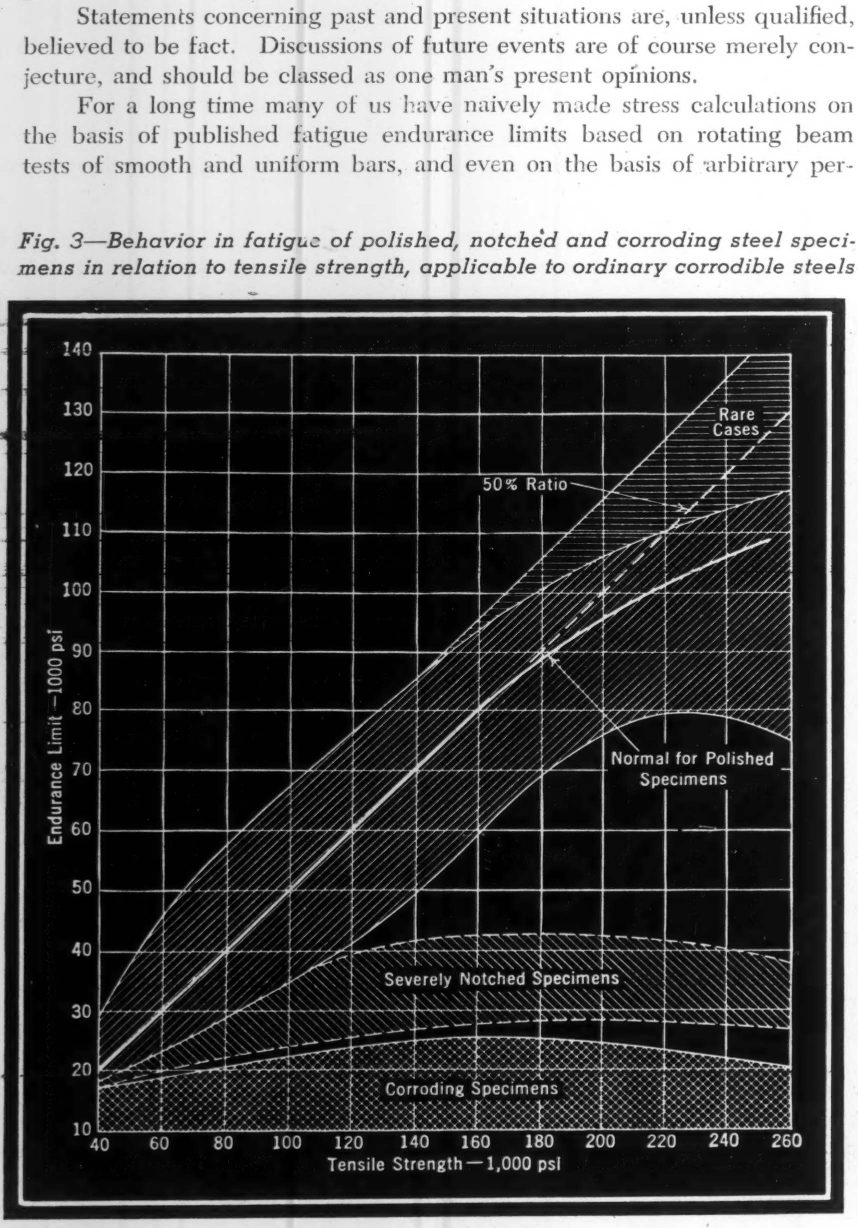
Class	Hardness: Rockwell	Hardness: Brinell Diameter	Hardness: Brinell Hardness	Tensile Strength (psi)
1	B93- 99	4.40-4.00	187-229	92,000-110,000
2	B96-102	4.20-3.85	207-248	100,000-118,000
3	C18- 26	4.10-3.75	217-262	103,000-125,000
4	C20- 28	4.00-3.70	229-269	110,000-128,000
5	C23- 30	3.90-3.60	241-285	115,000-135,000
6	C25- 32	3.80-3.50	255-302	121,000-143,000
7	C28- 35	3.70-3.40	269-321	128,000-152,000
8	C30- 36	3.60-3.35	285-331	135,000-157,000
9	C32- 38	3.50-3.25	302-352	143,000-168,000
10	C35- 40	3.40-3.15	321-375	152,000-180,000
11	C37- 42	3.25-3.05	352-401	167,000-196,000
12	C40- 45	3.15-2.95	375-429	180,000-214,000
13	C42- 46	3.05-2.90	401-444	196,000-223,000
14	C45- 49	2.95-2.80	429-477	214,000-247,000
15	C48- 52	2.85-2.70	461-514	235,000-275,000
16	C50- 54	2.75-2.65	495-534	261,000-291,000
17	C52- 56	2.70-2.60	514-555	
18	C55- 59	2.60-2.50	555-601	
19	C57- 61	2.55-2.45	578-627	
20	C60- 63			
21	C62- 65			

we shall then call normal economic conditions, and (c) the nature of those postwar "normal" conditions.

Statements concerning past and present situations are, unless qualified, believed to be fact. Discussions of future events are of course merely conjecture, and should be classed as one man's present opinions.

For a long time many of us have naively made stress calculations on the basis of published fatigue endurance limits based on rotating beam tests of smooth and uniform bars, and even on the basis of arbitrary percentages of ultimate tensile strengths. We have saved face by picking large factors of safety out of the air, and tacking them on our formulas. Engineers can't be blamed for this—they have been doing the best they can with the scanty information on behavior of metals furnished to them by metallurgists and mechanical researchers. As a result, we must rely on performance in simulated or actual service for our only reliable guidance.

Fig. 3—Behavior in fatigue of polished, notched and corroding steel specimens in relation to tensile strength, applicable to ordinary corrodible steels



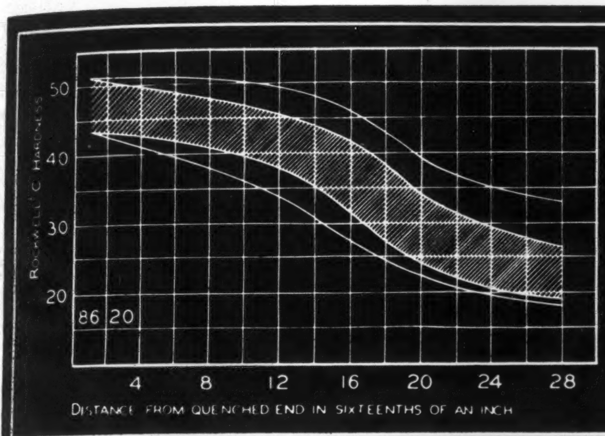
However, some light is beginning to glimmer through the fog. Certain trends and developments indicate progress is being made as result of long and intelligent studies of both fundamentals and specific applications.

It has been found, in certain parts free or practically free from stress concentrators, that fatigue endurance can be improved by increasing hardness far beyond limits previously thought practicable because of toughness limitations. In order to do this, at least one of two conceptions had to be revised in each case:

1. Toughness Is Essential. It might be difficult to call to mind a part which has been rejected solely because it was too tough. By negative inference the conclusion has been drawn that toughness is necessary in nearly every part. It nevertheless is true that some parts need little or no toughness in the steel. Whenever this is recognized, higher elastic limit and ultimate strength—at least near surfaces—and greater fatigue endurance may be utilized with resultant improved performance or weight reduction.

Is Deep Hardening Necessary?

2. Parts Must Be Hardened Throughout. We formerly specified (and still do, too frequently) high-alloy steels for highly stressed parts, hardened them throughout, and tempered them to as high hardnesses as were consistent with required toughnesses. If toughness really is required it often is possible, particularly if the part is to be subjected primarily to torsional or bending stresses, to use lower alloy or carbon steel so as to retain a tough, shock-resisting interior, and to harden only the outer highly stressed portions of the part to higher hardness than would be possible with deep-hardening steel, thus improving fatigue life at no sacrifice of overall toughness. Contrary to an idea held rather widely, the highest hardenability steel available may not be the



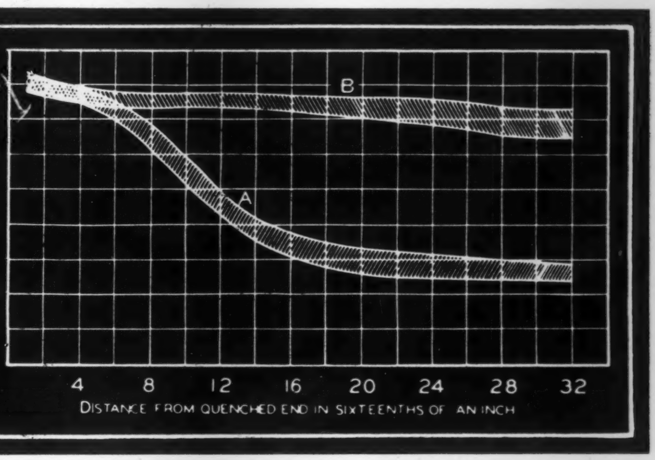


Fig. 4—Top left—End-quench hardenabilities of production heats of NE 8620 steel. Spread in 30 heats is shown by lowest and highest curves, desirable range for truck transmission gear application by shaded band. —*Courtesy E. O. Mann*

Fig. 5—Top right—End-quench hardenabilities of a steel treated with two different special addition agents

best, as has been repeatedly demonstrated in many production automotive and other applications, notably by Ford, Chrysler and Timken-Detroit Axle.

However, all parts cannot be handled this way. Some really must be hardened throughout. Unless range of repeated stress is reduced thereby, increasing hardness and strength beyond certain limits *throughout* such parts of irregular section may actually lower fatigue endurance, due to notch effect. As indicated in *Fig.* 3 (1)*, through-hardened parts having stress raisers (or subject to corrosion) can sometimes be improved in fatigue endurance by going in the opposite direction. First, the designer should try to remove sources of stress concentration. In cases where that is impossible, it frequently has been found helpful to lower the hardness of the part in order to permit some stress equalization through plastic flow. In this connection, greater knowledge concerning relative notch-sensitivity of materials is needed.

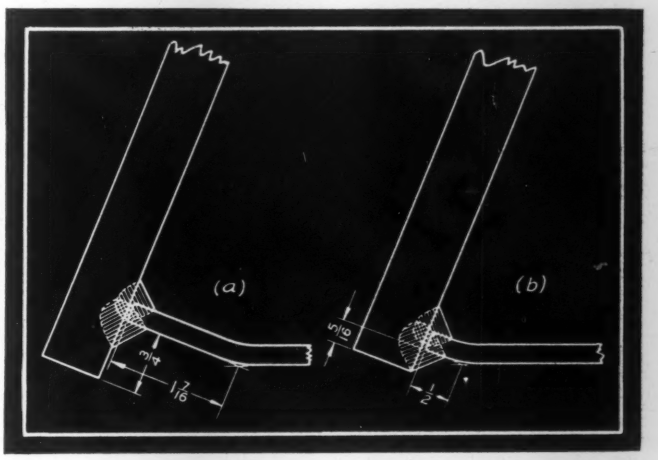
Improving Fatigue Strength of Parts

Another means of improving fatigue resistance is to improve surface finish. This is merely one way of eliminating stress raisers, and represents action which can be taken by engineer and production man. The value of smooth surfaces has long been appreciated, but cost considerations often prevent adoption except in the more efficient utilizations of materials such as in many aircraft and a few automotive parts.

It has been asserted by Almen (2) that fatigue failures occur only in tension. To increase fatigue endurance on this basis, initial compressive stress may be imparted to outer portions of parts to be highly and repeatedly stressed in tension. Such stresses can be introduced by shot peening, rolling, cold drawing, nitriding, some other types of casehardening, etc. Use of one of these processes, especially shot peening, may also eliminate necessity for expensive finishing operations, some of which, incidentally, may impart to surfaces of parts tensile stresses which actually tend to offset intended benefits of high surface finish. Also, a roughened surface such as produced by shot peening may in some cases improve performance because of its better retention of lubricant than in the case of a smooth surface.

Some future effects of these recent developments in improving fatigue endurance may be:

Fig. 6—Below—Under same welding conditions steels A (Fig. 5) welded satisfactorily in both joints, but steels B cracked in joint (a), were successful in joint (b)



1. Reduced alloy content in steels for many heat-treated parts in which deep hardening is found unnecessary or undesirable
2. More intense fight by designers against stress concentrators in highly stressed parts
3. Great increase in use of shot peening and other applications of compressive stress to surfaces, and lowered cost of finishing many highly stressed parts
4. Further study of notch sensitivity with relation to composition and structure, on which more precise data are needed.

Determination and study of transformation characteristics (during cooling from above the critical temperatures) of steels of various compositions has paid big dividends in revealing possibilities for improved heat-treating processes.

For example, principles and rules of annealing have been postulated more precisely than before (3). Pre-

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*References in parentheses are listed at end of article.

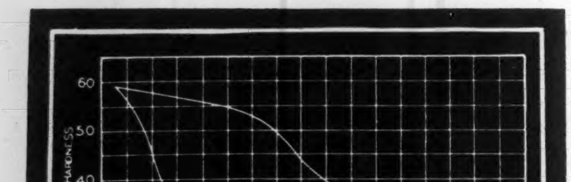
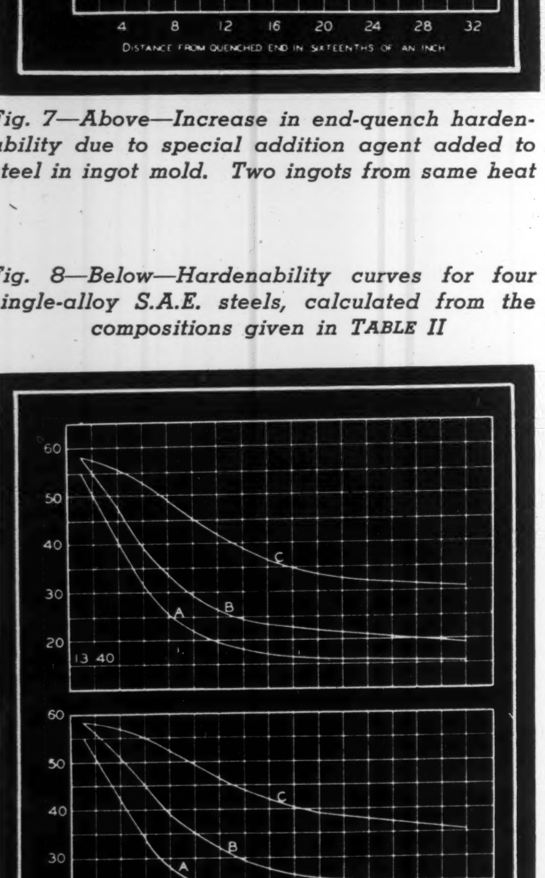


Fig. 7—Above—Increase in end-quench hardenability due to special addition agent added to steel in ingot mold. Two ingots from same heat

Fig. 8—Below—Hardenability curves for four single-alloy S.A.E. steels, calculated from the compositions given in TABLE II



cise annealing has helped to make possible the use of steel for cartridge cases (4), an application formerly considered impracticable. Examples of alloy steel parts, machinability of which has been improved by cycle annealing, are axle shafts, gears, tank track pins and end connectors, as well as tools of many kinds.

A process which permits application of thin hard cases to ordinary carbon and alloy steels without necessity of quenching and therefore with little distortion is carbon nitriding or dry cyaniding (5) (6). This has been applied to such parts as steel cylinder liners, oil pump shafts, valve parts, valve push rod ends, harmonic balancer pins, and (with oil quenching) high-carbon steel transmission gears.

Effects of these and other advances in heat-treating techniques will be lower manufacturing costs, better and more uniform products, and ability to use less critical and cheaper steels, both for war materiel made during the next period of peace and for industrial products.

Scientific study of foundry problems was started years ago by several large and progressive steel foundries. Developments have been broadened and accelerated during the war, largely because of demand for dependable cast armor and other important parts of military vehicles and artillery, where properties closely comparable with those of wrought steels were demanded.

Improvements have been made by more thorough application of known principles of steel melting and deoxidation, flow of molten metals, and cooling phenomena. All types of castings—static, centrifugal and pressure—have participated in the development.

Effects of improvements in steel castings will be more and diversified applications, especially for high-production parts of which expense for detailed study is warranted, with resultant decreases in amounts of metal required and in fabricating costs. Greater use of X-ray and gamma-ray inspection techniques will undoubtedly be made after the war than before. It also may be found that small jobbing foundries will experience greater difficulty in competing in production of high-quality castings, because of lack of men, facilities and money to permit detailed studies required to "work out the bugs" in production of intricate castings.

Art of Welding Becomes a Science

Great strides have been made during this war in the progress of welding from an art to a science. Two of the outstanding advances are spot welding and arc welding of armor and other highly stressed parts of carbon and alloy steels previously considered unweldable, at least for primary structures (*Fig.* 2). Intensive industry-wide cooperative studies of basic principles, designing and building new types of equipment, X-ray examination, ballistic tests, careful training of personnel, close supervision, and above all the ruling idea "it must be done" have been responsible for these developments.

Several new electrodes have been developed for better welding of sheet steel (*Fig.* 1) in all positions, with alternating current as well as direct current equipment. New materials for welding cast iron (7) and for hard surfacing have widened the scopes of those operations. Improvements in arc-welding machines also have been noteworthy.

Gas torch applications with respect to welding have broadened and improved. Flame cutting of steel has so increased in accuracy that many intricate parts are now being cut so nicely that no subsequent machining is required (8). Flame descaling of steel plate (9) and flame gouging of weld roots (8) are important aids to high-speed production.

Resistance-welding equipment and technique have advanced remarkably. By pulsating technique, spot welding of heavy-gage sections has been made possible (10). Stored energy equipment of several types has been developed; with this, rapid spot welding of aluminum and other metals as well as steel (11) is possible without the large power demands involved in using conventional alternating current machines. Use of synchronous timing equipment has made possible extremely accurate performance, with one stroke of the spot welder, of a sequence of operations consisting of weld, grain refine, quench, and temper (12) (13).

Expected effects of these and other welding developments in industry may be summed up in a few words: Vastly increased use of welding, especially arc and resistance, to replace more expensive assembling methods. All-welded ships, tanks and many other types of war materiel will lead the way to more welded road and rail transportation equipment as well as machinery and structures of countless types.

In order to bring this about, greater attention will be paid to "design for welding". It will not be sufficient merely to make sure welding operations are correctly specified; welding engineers must be allowed to influence thought of designers from the first conception of an assembly, long before detailing is begun.

Evaluation of Alloying Elements

Effect on steel of virtually all alloying elements is now expressed as increased hardenability. Absolute and relative effects of the several commonly used elements have been evaluated by several investigators, the method devised by Grossmann (14) being at present the most widely accepted. It now is realized that effect on hardenability of a single element often is exceeded by the cumulative effect of the same total percentage of several elements each present in small amounts.

In past years, steels often were selected for certain parts because of one or more of the following:

1. Sales efforts of alloy manufacturers
2. Preferences and recommendations of steel suppliers
3. Users' customs and prejudices (familiarity with certain steel types)
4. Lack of good heat-treating equipment and need for "foolproof" steels
5. Willingness of steel mill to furnish "residual" alloying elements, not specified or publicly acknowledged and not directly paid for.

Much debunking and self-education already have been accomplished. We realize that many low-alloy steels of equivalent carbon contents are interchangeable, and that carbon may sometimes be substituted for alloying elements, provided heat-treating characteristics are understood. Efforts are being made to express such interchangeability in tables and charts. Recommended practices have been established (15) for selection and substitution of British En steels for deep-hardening parts, on basis of usable sections for heat treatment to various physical properties, although those recommendations may be classed by some as ultra-conservative. A group of American metallurgists, headed by F. C. Young, working as a subcommittee of the S.A.E. War Engineering Board Iron and Steel committee, is attempting to classify steels on basis of performance requirements; progress is slow, because one of the chief products of discussion so far is realization that certain data which are considered essential have never been compiled.

Future effects of more precise evaluation of alloying elements would appear to be:

1. Use of Less Alloy. Gradual reduction in total amount of alloying elements used, with possible increase in complexity of alloy steels, is an expected trend.

2. More Intelligent Specification of Materials by Engineers. The trend may be toward stipulation of mechanical requirements, with no mention of steel to be used, leaving steel selection to the metallurgist. As an example, for parts to be hardened and tempered, a suggested list of mechanical property ranges is shown in Table I. The ranges given cover all requirements from soft to hard steels; there are probably more ranges than would be required by any one group of designers, but each may be classed as "commercial" with present-day standard steels and heat-treating equipment of average precision.

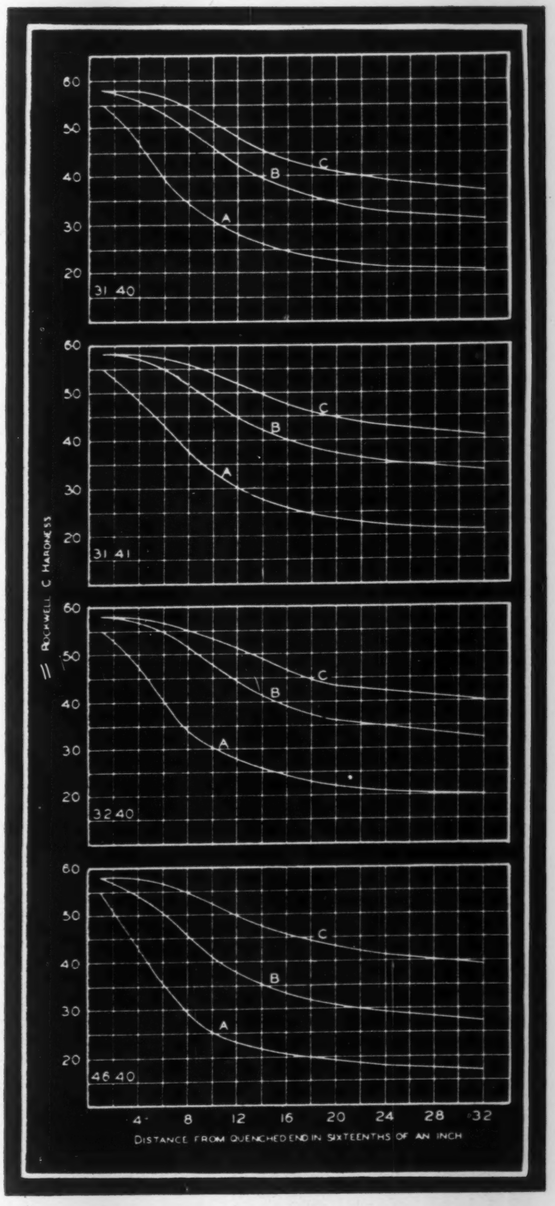


Fig. 9—Hardenability curves for two-alloy S.A.E. steels, calculated from compositions in Table II

3. More Intelligent and Economical Selection of Steels by Metallurgists. Selection will be based more upon physical and me-

TABLE II

Composition Limits of Some .40% Carbon Low-Alloy Steels*

		Chemical Composition (%)						
		Carbon	Manganese	Silicon	Nickel	Chromium	Molybdenum	Critical Diameter
1340	A	.38	1.60	.20	...	...	...	1.57
	B	.43	1.90	.35	...	...	...	2.12
	C	.43	1.90	.35	.25	.20	.06	4.03
2340	A	.38	.70	.20	3.25	...	...	1.81
	B	.43	.90	.35	3.75	...	...	2.74
	C	.43	.90	.35	3.75	.20	.06	4.75
4040	A	.38	.75	.20	...	...	.20	1.40
	B	.43	1.00	.35	...	...	.30	2.45
	C	.43	1.00	.35	.25	.20	.30	3.90
5140	A	.38	.70	.20	...	.70	...	1.91
	B	.43	.90	.35	...	.90	...	2.97
	C	.43	.90	.35	.25	.90	.06	3.83
3140	A	.38	.70	.20	1.10	.55	...	2.43
	B	.43	.90	.35	1.40	.75	...	4.13
	C	.43	.90	.35	1.40	.75	.06	4.92
3141	A	.38	.70	.20	1.10	.70	...	2.66
	B	.43	.90	.35	1.40	.90	...	4.47
	C	.43	.90	.35	1.40	.90	.06	5.33
3240	A	.38	.40	.20	1.65	.90	...	2.38
	B	.43	.60	.35	2.00	1.20	...	4.38
	C	.43	.60	.35	2.00	1.20	.06	5.22
4640	A	.38	.60	.20	1.65	...	.20	1.93
	B	.43	.80	.35	2.00	...	.30	3.54
	C	.43	.80	.35	2.00	.20	.30	5.15
8640	A	.38	.75	.20	.40	.40	.15	2.72
	B	.43	1.00	.35	.70	.60	.25	6.08
	C	.43	1.00	.35	.70	.60	.25	6.08
9440	A	.38	0.90	.40	.20	.20	.08	2.32
	B	.43	1.20	.60	.50	.40	.15	5.28
	C	.43	1.20	.60	.50	.40	.15	5.28

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*For each steel "A" represents all elements at low limits, "B" all specified elements at high limit with no unspecified elements present, "C" all specified elements at high limit plus maximum permissible amounts of unspecified nickel, chromium and molybdenum. "Critical Diameter" means maximum diameter in which each steel will harden, with ideal quench, to 50 per cent martensite structure at center, calculated by Grossmann's method.

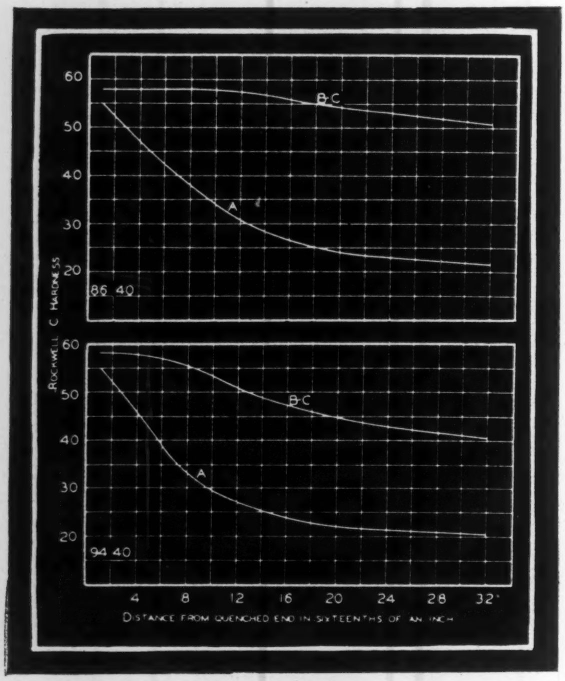


Fig. 10—Hardenability curves for three-alloy N.E. steels, calculated from the compositions listed in TABLE II

chanical requirements than directly upon chemical composition. However, for some applications relative susceptibilities of hardenability and maximum hardness to prior treatment and structure may be considered. A frank and intelligent discussion of steel selection has been made by Gillett and others in a War Metallurgy committee report (16).

Hardenability is one metallurgical term which should be understandable even to the most aloof engineer. It is not a foreign-language word, nor coined in honor of someone; it means simply the ability to harden (by heating and quenching).

Factors Controlling Hardenability

Steels must of course be compounded by adding chemical ingredients. In structural steels to be heat treated, however, chemical compositions are but means to ends, the most important end usually being proper hardening. Unfortunately, composition as ordinarily determined is not the sole criterion of hardenability; grain size and other factors, some difficult to evaluate, also are influential. After years of working on several of the factors, metallurgists finally began to realize that it might be simpler and more satisfactory to measure what was wanted—hardenability—than for users to continue to try to control a set of factors growing gradually more complicated and reaching far back into the steel's history.

Hardenability has been measured in many ways, to control steels and heat treatments for specific parts. The procedure most generally accepted is the end-quench or so-called Jominy method, developed by Jominy and Boegehold. The method consists essentially of quenching with a jet of water one end only of a uniformly heated test bar until the entire bar is cold, then measuring hardness along the side of the bar and plotting hardness readings; all details of the method have been carefully standardized (17). Hardness adjacent to quenched end is indicative of maximum surface hardness obtainable, and shape of curve evaluates hardenability of the piece of steel tested. In general, hardenability of a well-made heat of steel is reasonably uniform throughout the heat.

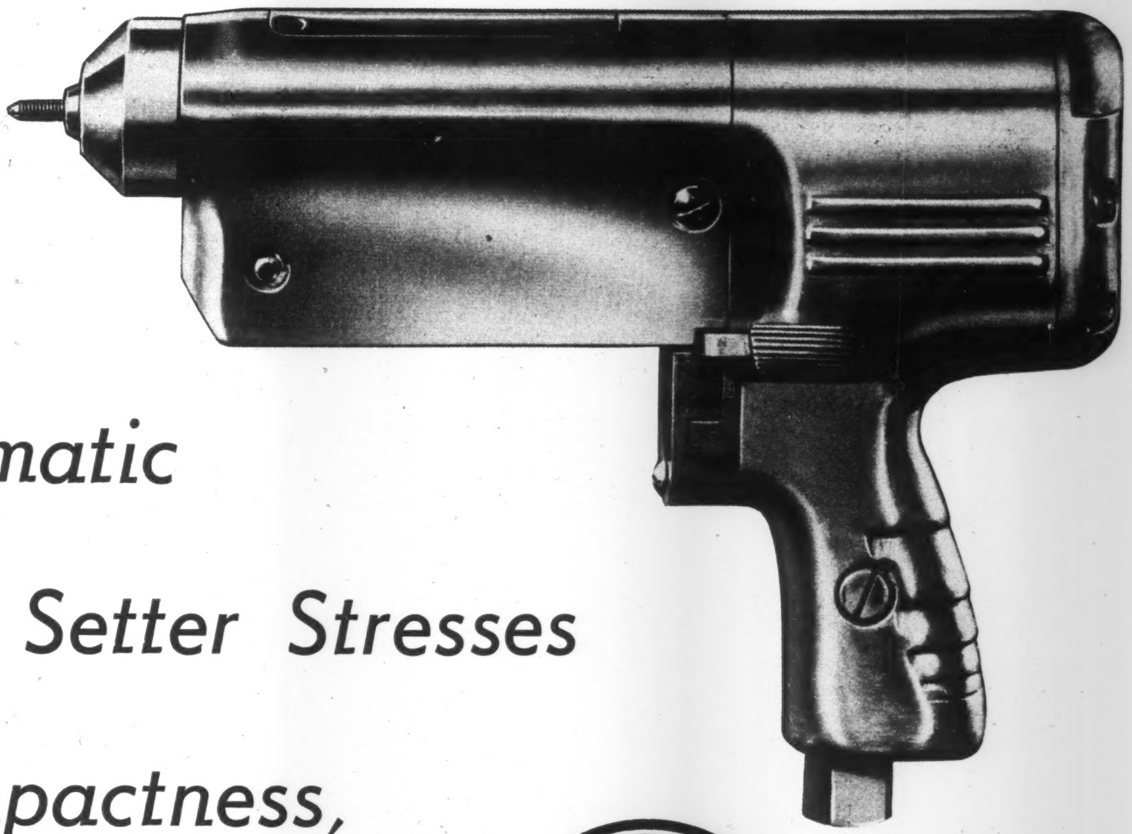
Hardenability of conventionally made steels can be calculated from composition and grain size, using methods devised by Grossmann (14) and Field (18). Results appear fairly accurate for steels of medium hardenability, somewhat less so for steels of very low and very high hardenability.

In 1942 suggestions were made by some steel producers that users specify hardenabilities and general types of steel desired, permitting producers to vary compositions beyond standard limits provided hardenability requirements were met. We were outgrowing the conception that hardenability of a standard steel type could be represented by a single-line curve, but bands representing spreads of hardenability for standard composition ranges had not been established. Therefore, no one knew (for publication, at least) what ranges in hardenability steelmakers could reasonably be expected to meet.

Spread in hardenability within a given steel type is determined by:

1. Variations in contents of specified elements
2. Variations in contents of residual or unspecified elements

(*Continued on Page* 162)



Pneumatic Rivet Setter Stresses Compactness, Light Weight

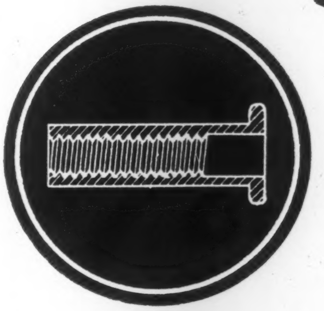


Fig. 1—Unit construction combined with maximum automaticity make this hand tool an outstanding design accomplishment. At left is shown a cross section of special rivet

By W. G. Mitchell

Chief Engineer
Independent Pneumatic Tool Co.

WHEN the "rivnut", a new type of blind rivet, was developed, it became necessary to furnish a tool with which to set it. Since this tool was to be handled by unskilled personnel its operation had to be as simple, automatic and foolproof as possible. Also, because it was to be a hand tool, it had to be sturdy enough to withstand normal abuse, yet light in weight so as not to incur undue worker fatigue.

As pictured in *Fig.* 1, the rivnut is of subtantially the same form as the customary flat-head or countersunk-head rivet but has a hole through its center, one end of which is threaded. Upsetting it entails threading a mandrel into the hole, holding the head stationary and pulling back on the mandrel with a force sufficient to make the shank of the rivet bulge out or upset.

In designing the rivnut setting tool the following requirements had to be met: Accurate control of the amount of set given the rivets; the mandrel must not rotate while the rivet is being set; positive cycle of operations; positive indication that full set has been given the rivet when the upset portion cannot be seen.

As finally developed, *Fig.* 2, the tool is an interesting example of a light-weight, compact, portable power-driven unit which performs the necessary series of operations in unalterable sequence and indicates the satisfactory attainment of the desired results. It consists of four major assemblies:

1. Air motor and gear reduction
2. Rivet-setting assembly consisting of air cylinder, cam, roller and plunger parts
3. Indicating assembly, consisting of contacts, battery, wiring and lamp
4. Control mechanism, consisting of valve, trigger unit and air channels.

From an external point of view the machine consists of two main housings divided just forward of the handle. Since the operating end of the mandrel can be no larger than the internal diameter of the rivet, this is the most vulnerable part of the tool and about the only one requiring replacement except after particularly long service. The exploded view, *Fig.* 4, illustrates the ready accessi-

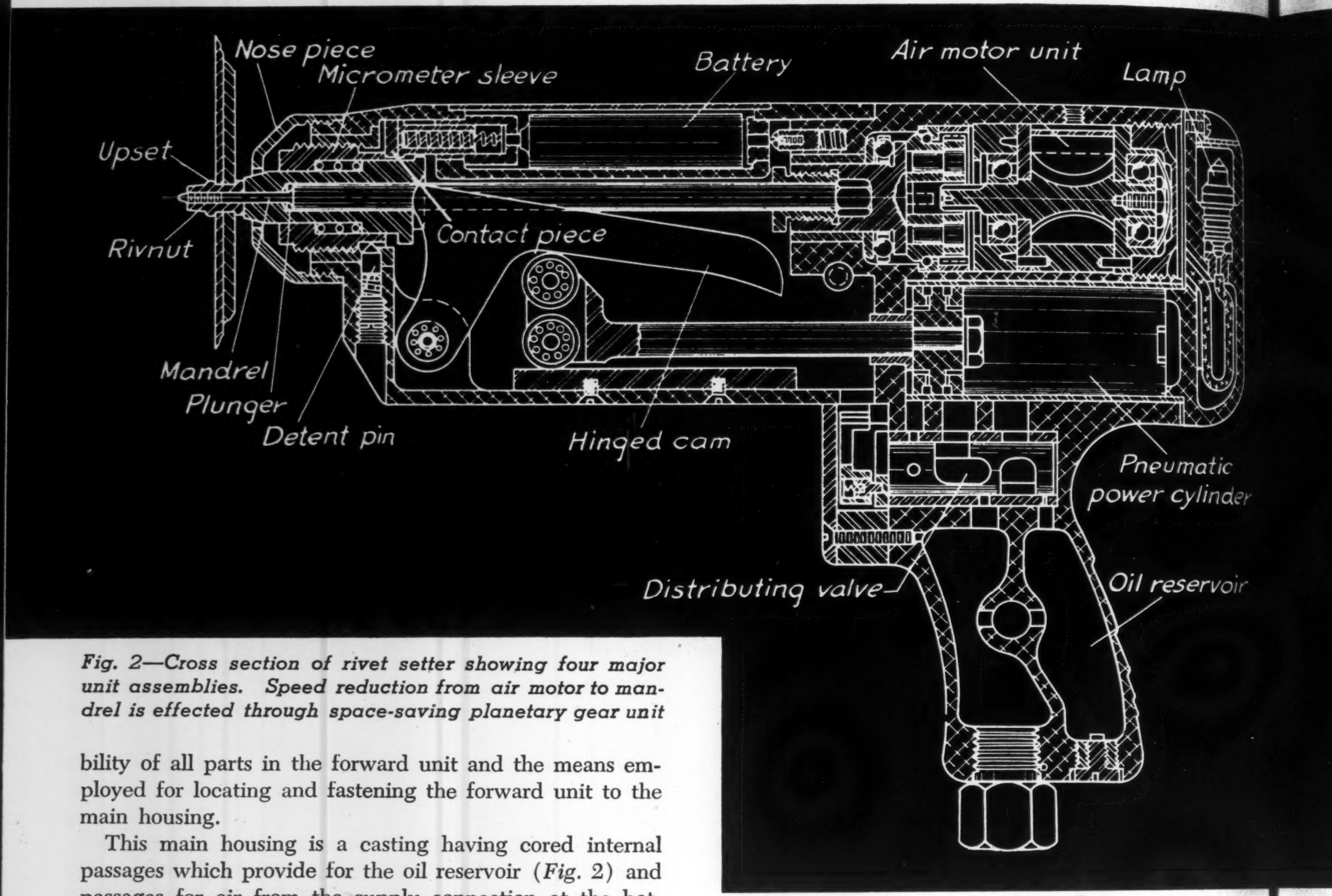


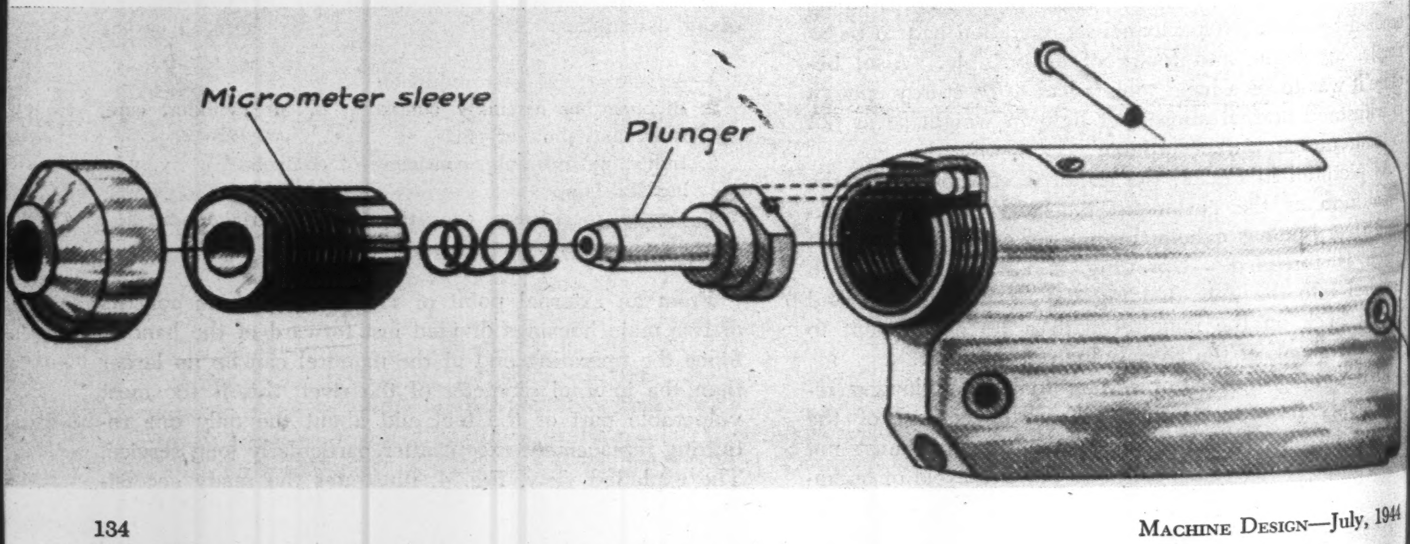
Fig. 2—Cross section of rivet setter showing four major unit assemblies. Speed reduction from air motor to mandrel is effected through space-saving planetary gear unit

bility of all parts in the forward unit and the means employed for locating and fastening the forward unit to the main housing.

This main housing is a casting having cored internal passages which provide for the oil reservoir (*Fig.* 2) and passages for air from the supply connection at the bottom of the handle through the distributing valve to the reversible air motor in the upper righthand portion and to the pneumatic power cylinder just below it. The valve is a cylinder with appropriate openings and channels. It is revolved through 90 degrees by each stroke of the operating or control lever.

Both from a production and service point of view the unit plan of construction outlined above has proved extremely advantageous, accessibility being of a high order. The air motor unit is assembled into a machined cylindrical bore in the upper portion of the main housing. Another bore directly below it is fitted with a steel liner with the requisite openings to house the pneumatic cylinder. Both of these bores are covered with a cast cap which houses the indicating lamp. The pair of rollers moving between the hardened steel base plate and the hinged cam are positively spaced and floatingly attached to the pneumatic piston rod to prevent any possibility of rod misalignment. The hinged cam has a curvature designed to produce ample force on the plunger in all angular positions in accordance with experimental determinations as to the power required. Since the mandrel is stationary during the cycle when the plunger moves forward it is in tension only, as a result of the compressional force exerted on the rivet, and is not under combined torsion and tension as was the case in previous designs.

Necessary adjustments on a hand tool of this type must



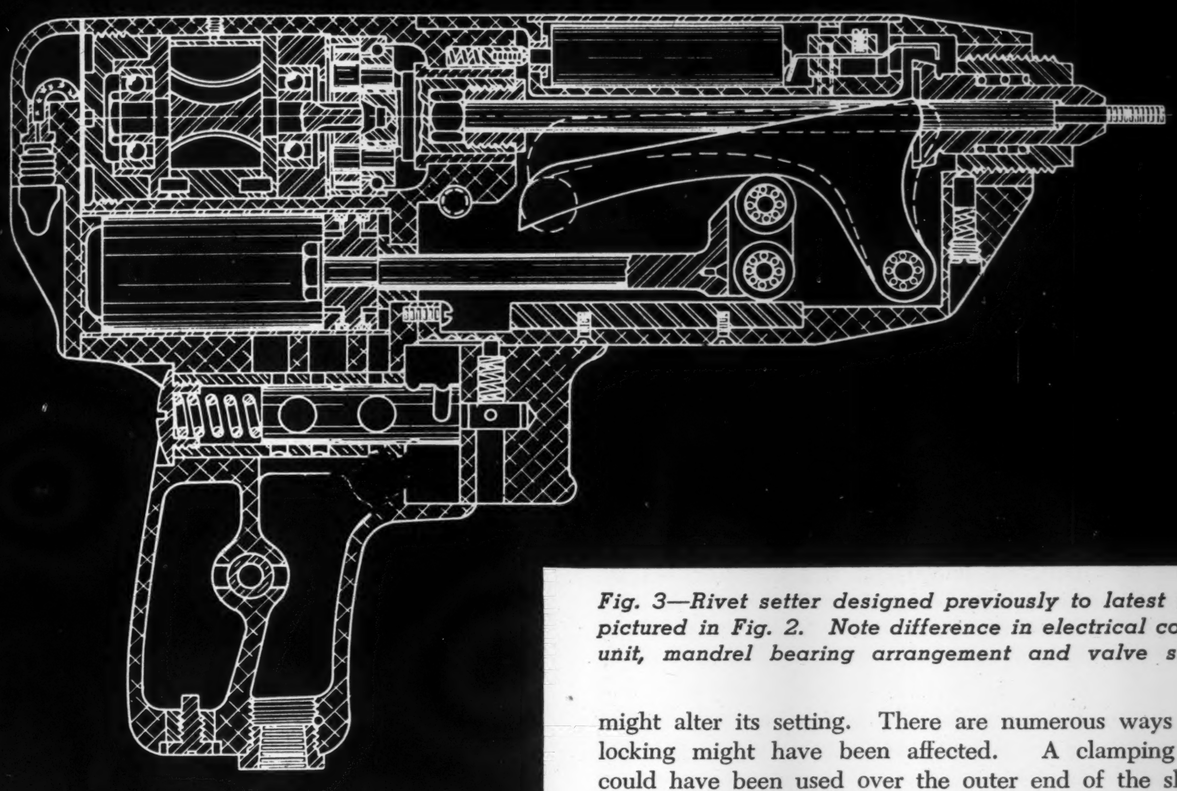


Fig. 3—Rivet setter designed previously to latest mode pictured in Fig. 2. Note difference in electrical contact unit, mandrel bearing arrangement and valve sleeve

be as few in number as possible and should be capable of being made at maximum speed and convenience. Only one such adjustment is required on the tool under discussion and that is the setting of the micrometer sleeve (see *Fig.* 2). It will be seen that the movement of the plunger through the micrometer sleeve is stopped when its flange strikes the inner end of the sleeve. Thus, by threading the sleeve in or out of the housing nose, the length of stroke permitted the plunger is controlled. (Different stroke lengths are required for various types of rivnuts.)

Obviously, once the micrometer sleeve has been set it must be locked, otherwise normal handling and operating might alter its setting. There are numerous ways such locking might have been affected. A clamping ring could have been used over the outer end of the sleeve, being tightened with three screws. A jam nut also would have worked. Again, a setscrew might have been employed. However, all of these devices would have required tightening with screwdriver or wrench after the setting was made, without the benefit of the means of measuring the amount of change provided by the clicks of a detent.

The manner in which locking actually is accomplished is extremely simple and automatically locks the sleeve in whatever position it is set. As shown in *Fig.* 4, the micrometer sleeve carries a .031-inch pitch thread (thirty-two threads per inch) on its outside diameter and in addition eighteen equally spaced grooves. Thus, as the sleeve is turned within its bushing, the detent pin shown in *Fig.* 2 will snap in and out of the grooves and effectively hold the sleeve against turning when it is engaged with any of the grooves. This means of course that the lengthwise setting of the sleeve can be locked at equal intervals of slightly under .002-inch.

It is necessary to indicate to the operator in some manner that the rivnut has been set properly at the completion of the second of the four operations which constitute the complete cycle. In the model immediately preced-

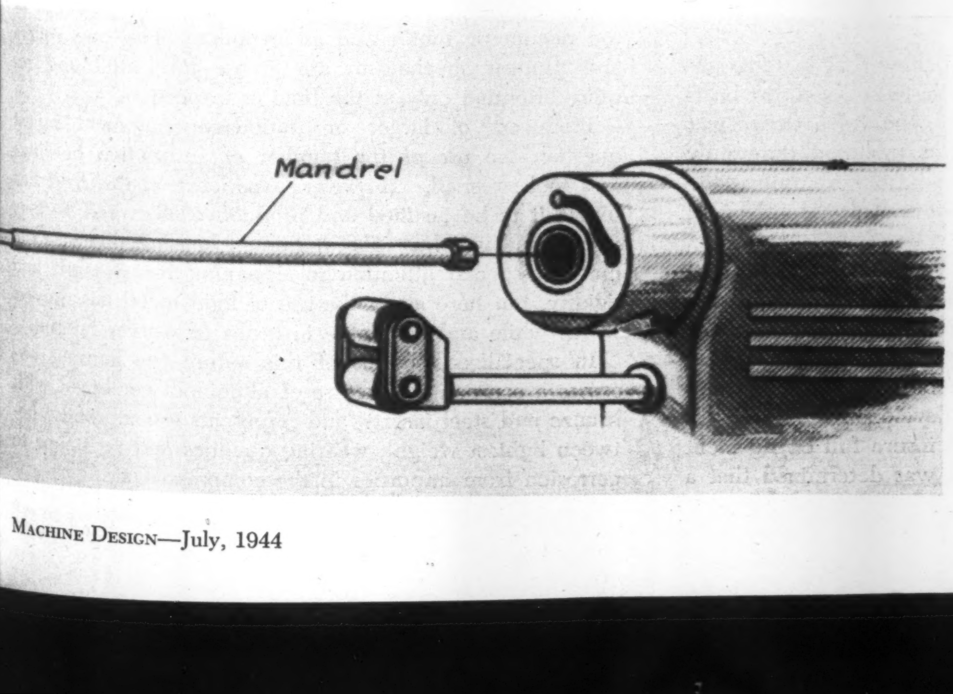


Fig. 4—Exploded view shows ready accessibility of parts in the forward unit and sequence of their assembly

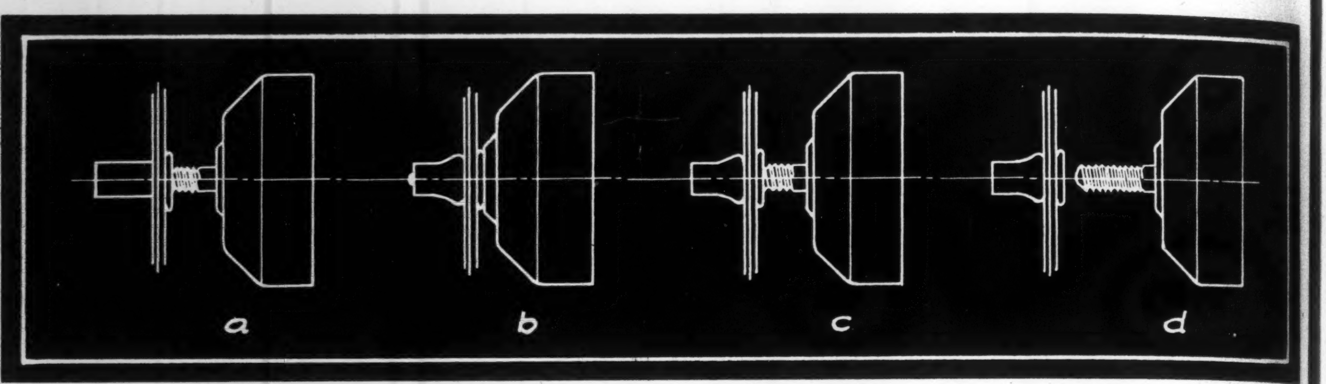


Fig. 5—Demonstrating the four operating cycles of the rivet setter. After the mandrel screws into the rivet as at a, the plunger moves out to upset its shank (view b). Next, as in view c, the mandrel reverses direction and unscrews from the rivet. During fourth cycle, view d, all motion stops and tool completes driving cycle

ing the one under discussion this was accomplished by means of the formed strip contact which is shown in *Fig.* 3 mounted between two blocks immediately ahead of the battery. The end of this contact strip was positioned at assembly to touch the tapered outside diameter of the plunger flange at the instant the plunger had been urged far enough forward to upset the rivnut fully. This design worked well enough but required resetting for each size of rivnut and this resetting had to be effected independently of the resetting of the plunger stroke length.

New Design Obviates Contact Setting

This inconvenience has been eliminated in the present design shown in *Fig.* 2. Here the flange on the internal end of the plunger is offset at the top to provide the same amount of space between its inner surface and the contact point of the electrical indicating device, as the space that exists between the inner face of its bottom flange and the stop face of the micrometer sleeve. By this means electrical contact is established whenever the plunger makes its full stroke. Since the contact piece is spring loaded against the inside face of the micrometer sleeve it naturally will follow the linear motion of the sleeve, obviating the necessity for resetting the contact each time the micrometer sleeve is readjusted. The battery, housed directly above the cam, supplies the current to light the indicating lamp at the rear whenever contact is established.

Reference to *Figs.* 2 and 3 will show that in redesigning the unit it was deemed advisable to add a thrust bearing to the mandrel drive assembly, the valve sleeve was improved, the indicating lamp was mounted differently and a nose piece was added at the front to protect the micrometer sleeve adjustment and prevent unauthorized changes in adjustment.

Throughout the design of this machine, every effort was made to keep its weight and size down to a bare minimum. For example, while it was necessary to make the air motor rotor as small as possible, doing so meant inserting a speed reduction unit between its shaft and the mandrel because the torque developed by this high-speed motor was not adequate to insure full engagement of the mandrel in the rivnuts. It was determined that a speed reduction of 6¾ to 1 would result in sufficient torque. While this reduction could have been effected through the use of compound spur gears, such a design would have required more space than the planetary reduction unit which is employed. Since a 6¾ to 1 reduction was necessary, the fixed internal gear was made 5¾ times the diameter of the sun pinion (cut integrally on the air rotor shaft end), the planet pinions running on studs pressed into the driven member.

The hinged cam must exert a pressure of approximately 1000 pounds on the micrometer sleeve, and was economically produced to give maximum strength from forged and heat treated SAE 3140. The cam roller plate, likewise highly stressed, was made from SAE 3140 bar stock. Due to function and shape requirements, the piston rod was forged from SAE 6145 steel. With the hexagonal drive nut made integral with the shank, the mandrel is turned from SAE 6145 steel and its thread is milled carefully to exacting limits to insure free turning within the rivnut. The pneumatic rotor is made of SAE 4620 while the rotor bushing is NE 8620. The plunger is of SAE 3120. Since moisture usually is found in compressed air, the valve was machined from stainless steel, finish ground and surrounded by a bronze bushing. Both of the external housings are of cast aluminum alloy.

Since the cycle of the machine cannot be altered, the mandrel cannot enter the rivnut except on the first cycle when it is rotating in a clockwise direction. The mandrel is stationary during the second or squeezing cycle; thus it is impossible to remove it from the rivet until the third or counterclockwise cycle.

Fine oil-spray lubrication is provided each cycle from the reservoir through the control valve and passages to the pneumatic motor and air cylinder. The cam rollers and plunger mechanism are grease lubricated and require attention only at the time of inspection.

Producers of larger or stationary equipment might question the use of the number of antifriction bearings in a tool so small. However, experience in this field has shown it to be justified and productive of overall savings in size and weight. The aluminum housings likewise demanded special attention in designing for strength and rigidity, but here again the use of light metal has proved worth while and justifiable in terms of worker fatigue.

In specialized designs of this nature the housing becomes a composite structure of aluminum combined with bronze and steel inserts, and represents a compromise between lightest weight, wearing qualities and resistance to corrosion from impurties in the compressed air.

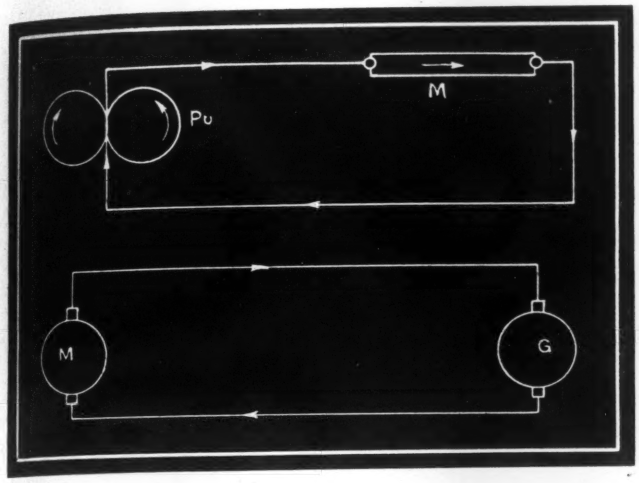


Fig. 1—Left—Direct-current electrical circuit and equivalent hydraulic circuit with power flow in ane direction

Fig. 2—Below—Hydraulic or pneumatic generator corresponding to alternating-current flow for electricity

Fig. 3—Bottom—Single-phase "alternating-current" hydraulic system

Fluid and Electrical Analogies Work Both Ways

By R. S. Elberty

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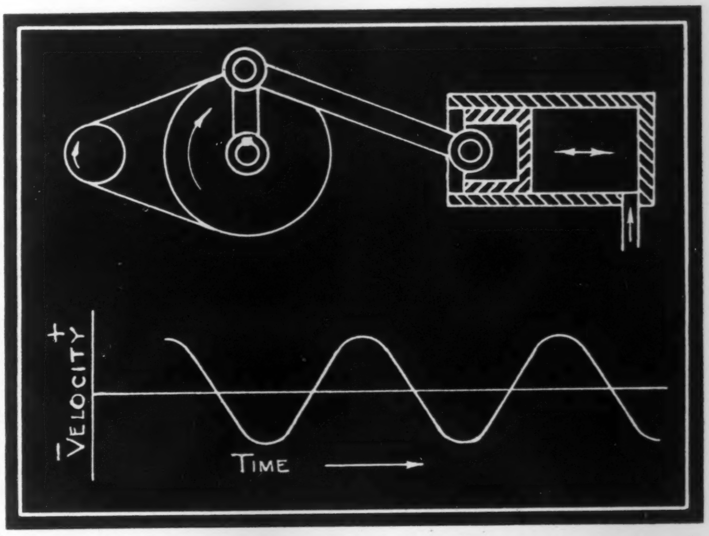
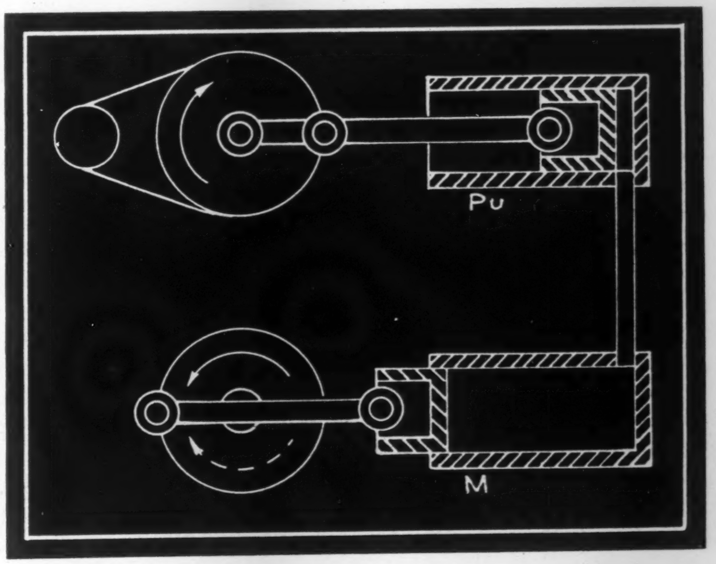




SCIENTISTS periodically try—but without success—to describe or define electricity in understandable terms. However, electricity is employed as a form of energy bearing a close relationship to fluid pressure. Pressure, rate of flow and resistance to flow are used in design calculations; and fluid circuits may be devised to illustrate electrical inductance and capacitance. On the other hand, this very resemblance points the way toward the development of new hydraulic and pneumatic devices to utilize principles developed in the application of electrical energy. To investigate these possibilities, we must understand the various words used to define similar qualities of each medium. For this purpose TABLE I shows the corresponding electrical and fluid definitions usually employed.

An electric circuit with a generator and a motor is shown in *Fig.* 1. This is a direct-current circuit, the power flow being continuous in one direction. The equivalent hydraulic circuit, also shown in *Fig.* 1, has a rotating pump and a reciprocating motor. Now there is one great difference between the two systems. The conventional electrical system has *constant* pressure and *variable* rate of flow, the hydraulic system has *constant* rate of flow and *variable* pressure. If the load in the electrical motor is increased the voltage remains substantially constant but the amperes increase. An increase in the load on the hydraulic motor will increase the pressure, the gallons per minute from the pump remaining constant. This is a fundamental difference. The great majority of electrical drives are constant pressure, but most hydraulic drives must be converted from constant-volume to constant-pressure systems by additional equipment or by complex constant-pressure pumps.

Another difference between electric and fluid circuits is in the resistance to flow. Electrical resistance is considered to be a constant in the circuit. The drop in pressure in an electric circuit then varies directly as the rate of flow (amperes) in the circuit (Ohm's law), but the flow of fluids is not nearly as simple. TABLE II shows how the pressure drop of fluids varies as the amount conducted is varied. This table does not consider turbulence, and the pressure drop will be greater under such a condition.

If electricity is considered as a fluid, it acts as a gas or elastic fluid since it fills the container. A capacitor is a container for electricity, and the amount of charge stored will vary as the size of the capacitor and the voltage or pressure. In like manner, the amount of air stored in a supply tank will vary as the size of the tank and the pressure.

Alternating-Current Systems

So far, this discussion has not considered alternating current. *Fig.* 2 illustrates a piston pump without valves. This pump might be considered filled with a liquid which is alternately pumped from the cylinder and drawn back as the piston reciprocates. This flow, as shown on the chart, represents a hydraulic equivalent of the single-phase alternating-current circuit. The difference in the analogy is that pressure remains constant in the electric circuit and volume varies as load increases; in the hydraulic circuit pressure will depend on the load, and volume is determined by displacement of the pump.

TABLE I

Corresponding Electrical and Fluid Terms

General Unit	Electrical Unit	Fluid Unit
Pressure	Volt	Pounds per Square Inch
Flow Rate	Ampere	Gallons per Minute
Work Rate	Watts	Horsepower
Total Work	Watthour	Horsepower-Hour

The pump of *Fig.* 2 is shown in *Fig.* 3 connected to a similar device to be used as a motor. The motor is shown on dead center but it has a flywheel to carry it over this position. It will run in either direction once it is started. This diagram illustrates the reason single-phase induction and synchronous motors must have starting devices.

In *Fig.* 4, the pump has two cylinders spaced at 90 degrees around the same crank making a difference in time between peak values from the two pumps. In electrical terms this is called a difference in phase. The phase angle being 90 degrees in this case, the pump is analagous to a two-phase electric generator. If there were three cylinders spaced at 120 degrees, the hydraulic machine could be compared to a three-phase generator.

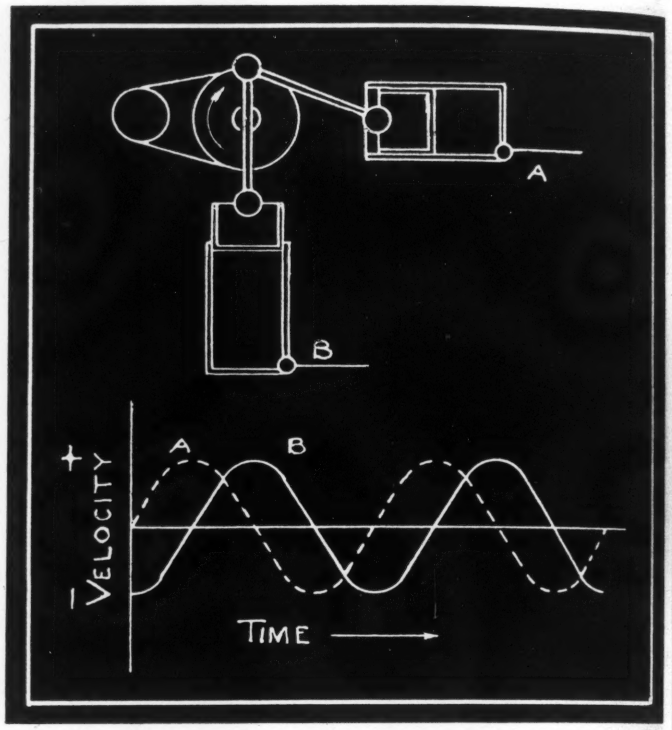
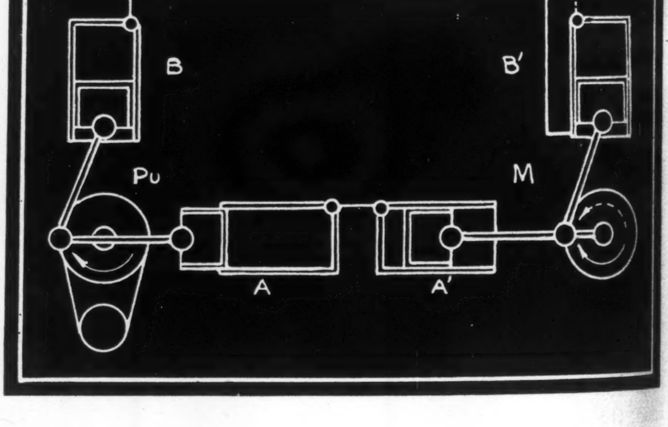


Fig. 4 — Above — Two-phase "alternating-current" hydraulic pumps. Chart shows relationship such as exists in two-phase electrical circuits

Fig. 5—Below—Two-phase alternating-current hydraulic circuit with pump and motor. Direction of rotation of the motor may be reversed by reversing the connections to one cylinder



Polyphase motors have starting characteristics which can be explained from *Fig.* 5. A two-phase pump and a two-phase motor are connected. There is no dead center position since one cylinder of the pump is always driving one cylinder of the motor. The motor must rotate in the direction shown. To reverse the rotation of the motor, the piping line might be moved to the top cylinder *B′*. Likewise, in a two or three-phase electric motor, rotation is reversed by reversing connections to one phase of the system.

Another interesting consideration is that electrical engineers are often criticized by mechanical and hydraulic designers for not developing a simple alternating-current adjustable-speed drive. This has its counterpart in hydraulics. How can one arrange to vary the speed of the motor in *Fig.* 5 without varying the speed of the pump? With a solution to this problem, some electrical designer might develop the adjustable-speed alternating-current electric motor.

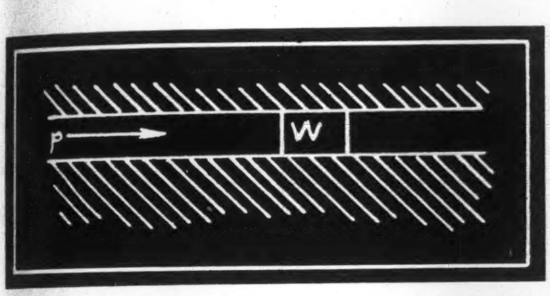
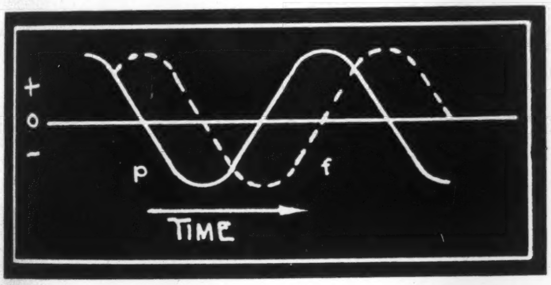


Fig. 6—Above—Pneumatic equivalent of electrical inductance

Fig. 7—Below—Effect of inertia on a pneumatic circuit shown graphically



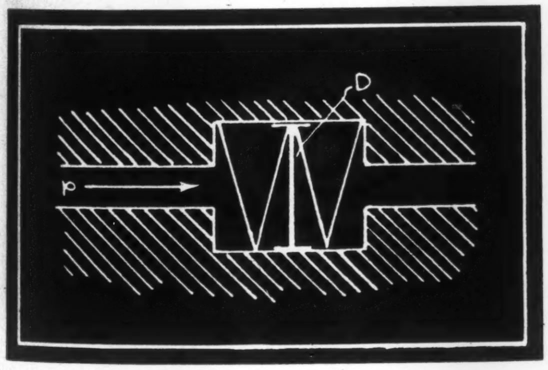
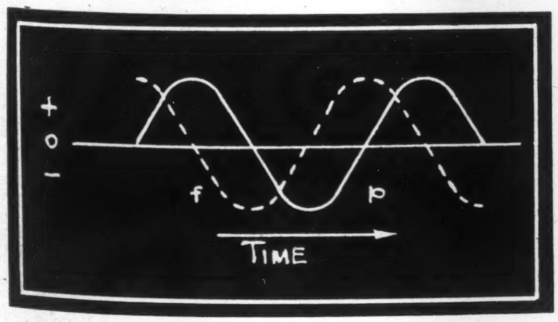


Fig. 8—Above—Electrical capacity illustrated by a pneumatic analogy

Fig. 9—Below—Addition of resistance to a pneumatic circuit causes the flow to lead the pressure



Electrical inductance is an inertia effect, a tendency to resist change. To work out a fluid equivalent, we must use a weightless gas fluid medium. *Fig.* 6 represents the equivalent of an inductance in a pneumatic circuit. The inertia member *W* must move in the conductor without friction. The friction will act as a resistance and will have the same effect as electrical resistance in an inductive circuit. In *Fig.* 6, if pressure *p* is rising, the weight *W* is being accelerated to the right, and the flow of the fluid is being retarded by the inertia action of the weight.

TABLE II

Relation Between Pressure Drop and Rate of Flow

Fluid	Pressure Drop in Conductor
Electricity	Varies directly as rate of flow
Air	Varies as square of rate of flow
Water	Varies as 1.86 power of rate of flow
Oil	Varies as cube of rate of flow

Now if the pressure *p* is diminished, or reversed, inertia of weight *W* tends to keep the flow of fluid moving. In *Fig.* 7, an alternating pressure is shown as *p* and the flow as *f*. The flow *lags* behind the pressure, and for pure inductance or inertia the angle of lag is 90 degrees. In electrical terms, power factor is the cosine of the phase angle between volts and amperes. Thus the analogous power factor for hydraulic systems is equal to the cosine of 90 degrees, or zero per cent lagging. For convenience the cosine is multiplied by 100 and referred to as "per cent".

Similarity of Capacity for Both Systems

Capacity in pneumatic circuits is shown in *Fig.* 8. The weightless piston *D* is spring supported without friction and seals the chamber so that the fluid will not leak past the edges. An increasing pressure *p* will cause the piston *D* to move to the right, storing energy in the springs. If *p* is reduced the springs move to the left, reversing the flow even though *p* might still be positive. In *Fig.* 9 the alternating pressure is shown as *p*, the flow as *f*. In this case, the flow *leads* the pressure, the angle of lead being 90 degrees, and the power factor is zero per cent leading.

In *Fig.* 6 a constant pressure will cause flow in the circuit, and direct current will flow through an electrical inductance for the same reason. In *Fig.* 8, a constant pressure will not cause a flow; likewise in electrical circuits direct current will not flow through a capacitor. If too much pressure is applied in *Fig.* 8, the springs break. Similarly if too much voltage is applied to a capacitor, the electrical insulating material breaks down.

Thus a useful method of attacking hydraulic or electrical problems is to employ understandable analogies. There are fluid analogies that will cover most of the basic principles of electricity, though there are many basic differences. Applying the reasoning of analogies the questions may be asked, can pneumatic devices be utilized to act the same as electrical devices or can an electric circuit be substituted for a hydraulic control or drive? The machine designer can broaden his scope of design by utilizing all forms of fluid energy, knowing that they have some basic points in common providing for a certain amount of interchangeability.

MACHINE *Editorial* DESIGN

Design's New Opportunity—Winning the Peace

WITH success of the European invasion assured and favorable news from other battle fronts, chief engineers and designers no longer have reason to feel that their primary efforts should be identified with design for war. That job has been done—and done well.

Already strong indications are emanating from Washington that are typical of the surge toward the setting up of plans for the future. As a result of the debacle at the Brewster Aeronautical plant caused by the sudden termination of plane contracts, steps are being taken to prevent such conditions arising as more and more war contracts are cut back or terminated. Further, WPB has authorized the purchase of machine tools for future, rather than war, needs; and Senator Walter F. George's committee is sponsoring a proposal to establish, among other things, an office of demobilization for war agencies and also to encourage construction projects to take care of unemployment conditions.

Though such unemployment may well arise after the war, it can be minimized considerably by the effectiveness with which designers have been able to apply themselves to postwar work thus far and the effort they put into the task in the short time that may remain. The situation is completely reversed from that of three years ago when we faced an inevitable war. Now the only question is how soon that war can be won.

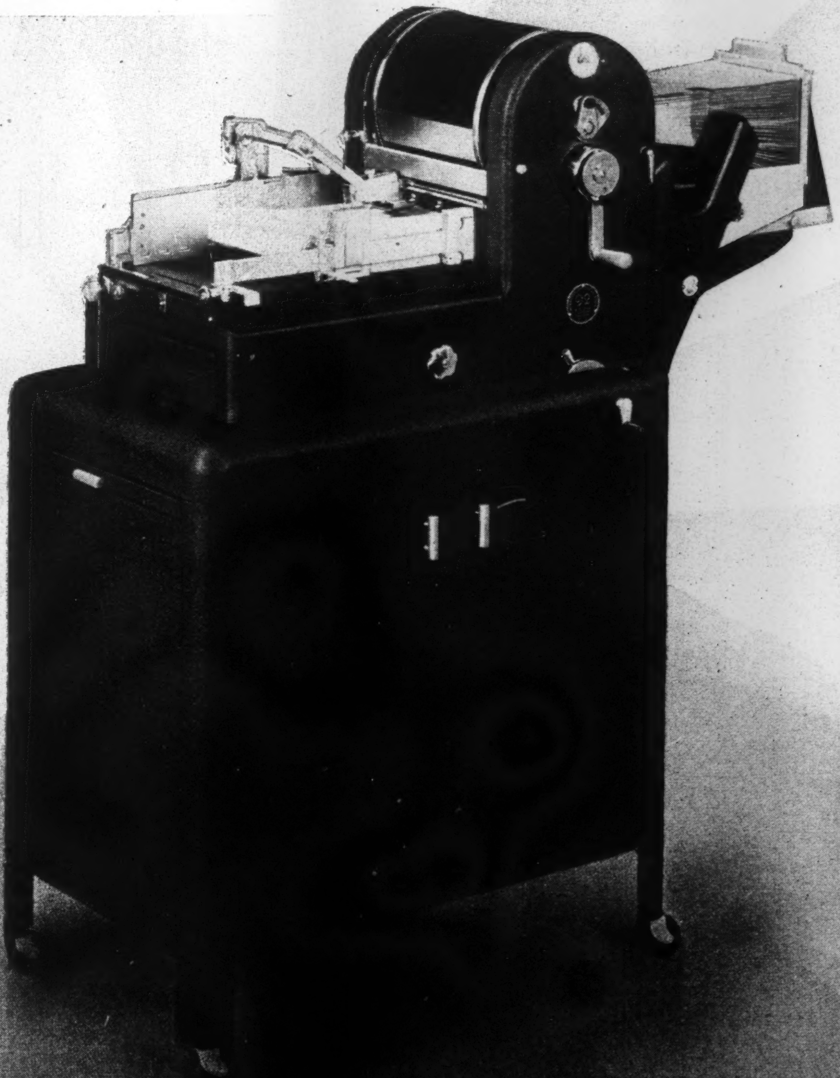
For designers still engaged in new military developments there still is much to be done—whether their designs eventually are employed in the war or for peace-time preparedness—a prime example being the utilization of the jet propulsion principle as discussed in the lead article in this issue and brought into new prominence by Germany's use of robot planes. But for all other designers whose war activities have ceased or are ceasing there now is an entirely different responsibility—the development of new machines that will go far toward "winning the peace".

L. E. Jermy

Outstanding Designs

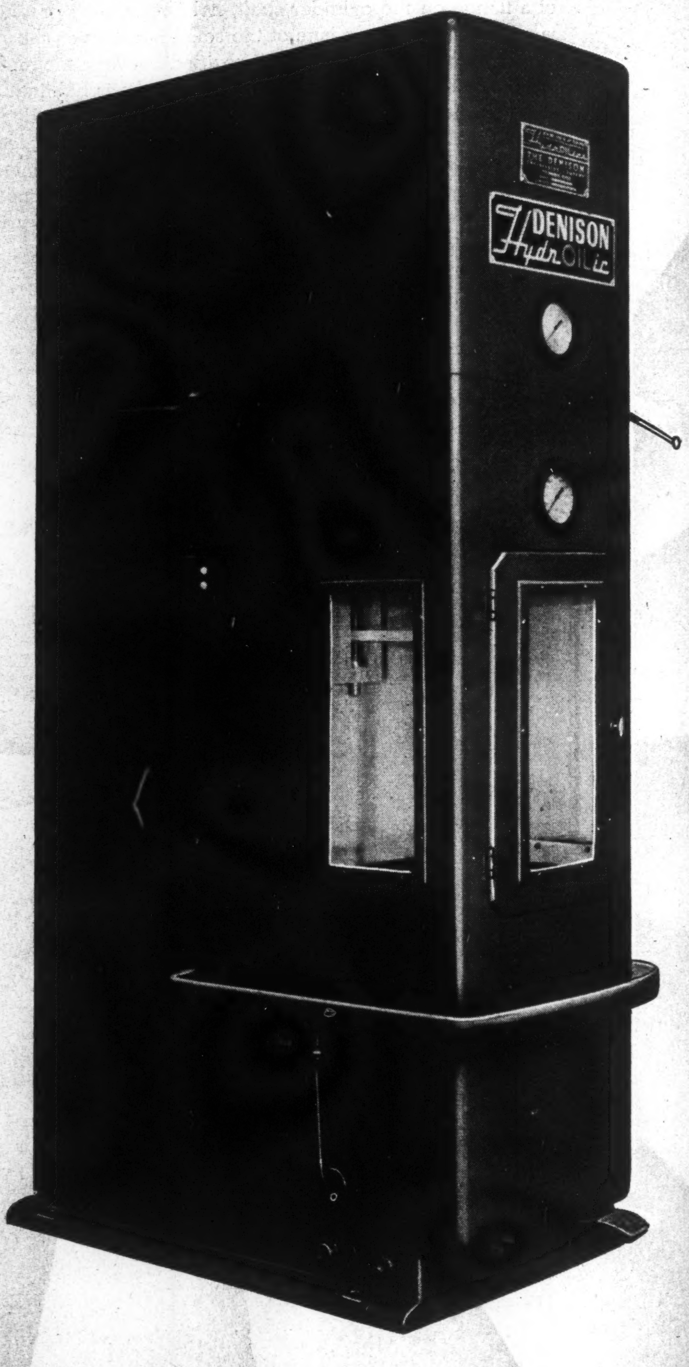
With main and unit housings made of formed sheet steel having rust-resisting wrinkle finish, and many of its control knobs and handles made of molded plastic, this automatic duplicating machine manufactured by A. B. Dick company represents truly modern design. It utilizes dozens of stamped and formed steel linkages, guides, cams, etc. Powered by an electric motor which is mounted resiliently on rubber pads, the machine's adjustable speed is obtained through use of a V-belt variable-speed unit of the spring-loaded sheave type. The small hand crank mounted at the top right-hand corner of the base cabinet is used to screw a rod into a block linked to a pivoted V-belt sheave mount; as the sheave mount is raised or lowered, the speed at which the cylinder is driven changes proportionately.

The brake for the cylinder consists of a thin strip spring wrapped part way around the diameter of a flange on the cylinder shaft, held at braking pressure by an auxiliary spring. Control of this brake is accomplished by means of the small hand lever shown immediately to the right of the cylinder crank. An interlocking mechanism is provided between the motor "on and off" lever and the brake lever so that the motor cannot be started while the brake is holding. The sheet pusher employs a corrugated rubber pad bonded to a sheet metal plate, and rubber-coated felt pads (spring loaded) are used for two of the paper guides. To prevent offset printing on adjacent sheets, blotting sheets are flipped between the printed sheets as they are ejected by the printing cylinder to the receiving tray. This is accomplished by two sets of formed fingers (one set on each side of sheets) which are actuated by pushrods working from the cycling mechanism. For adjusting the various guide plates, knurled thumb screws are used throughout.



Designed to test special tubing hydrostatically on a production basis at pressures as high as 6000 pounds per square inch, this press, manufactured by The Denison Engineering company, uses water as a pressure medium. Pressure is built up by forcing a ram into the water-filled tube after both of its ends have been sealed. Water is introduced into the tubing by means of the pedal control. The operator depresses the ram control handle and the ram lowers to seal the top end of the tubing (at the same time a fixture seals the tube bottom). Pressure then is increased on the ram and after a predetermined intensity has been reached, another ram inside of the main one intensifies the test pressure to the desired amount. Sealing pressure increases in proportion to the intensified test pressure and there is a definite relationship existing constantly between the two.

The intensifier ram is so designed that if there should be a slow leak in the tube it will complete its stroke and then drop pressure. This is done to prevent erroneous pressure reading if a leak is not observed. Power unit and all other hydraulic equipment is contained within the press and is accessible from the rear. Motor and pump are mounted as a unit for ease in servicing. The press frame is of welded steel construction having the oil reservoir welded in as part of the frame. With a honed and polished bore, the main cylinder is made of cast Meehanite. The main piston and ram are of steel and piston rings are used for sealing. The intensifier piston is made of bronze with bronze sealing rings. It is single acting, spring returned, and has an intensifying ratio of approximately three to one. Steel, hardened and ground, with polished chromium-plated surface is used for the intensifier ram, polished chromium being an ideal sealing surface. Interior of the test chamber is painted with white enamel and contains a shielded light for good visibility. The operator is protected from high-pressure splashing by a metal shield and transparent plastic windows.



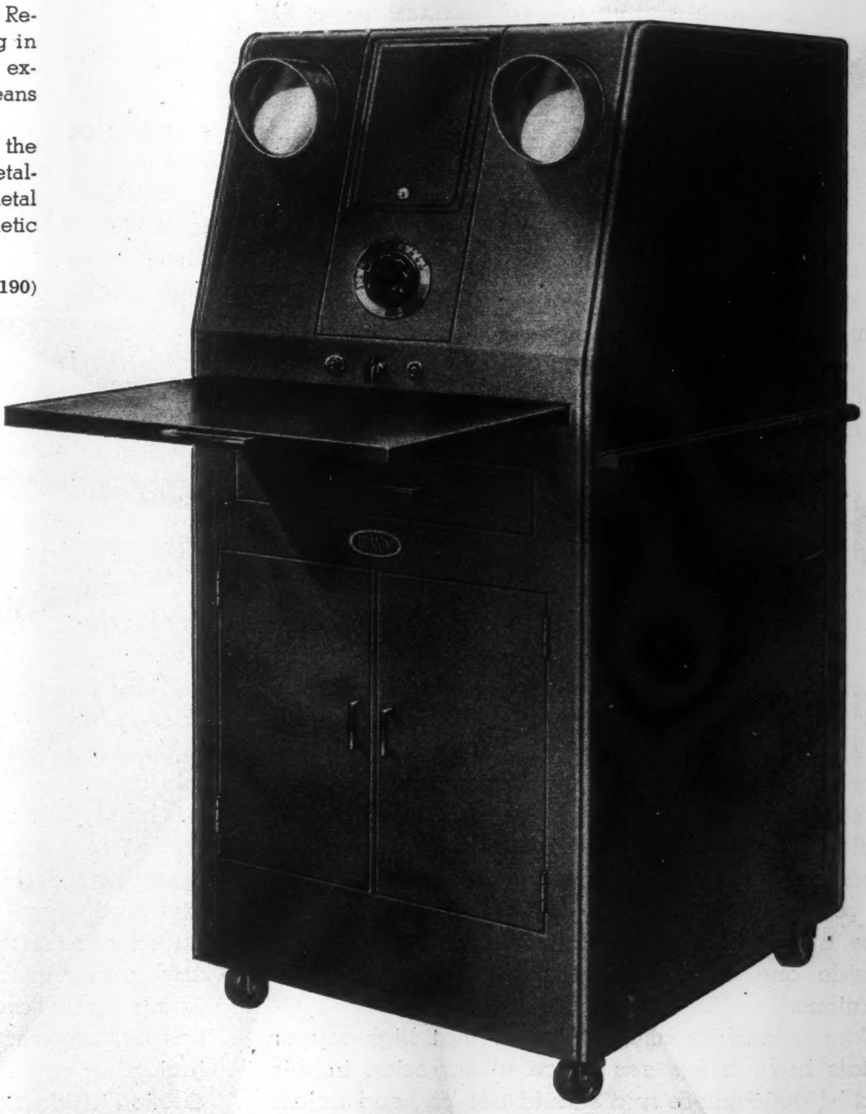
Utilizing the principle that the metallurgical properties of different metal parts cause variations in the core loss of a tuned pickup coil, this electronic testing machine, manufactured by Allen B. Du Mont Laboratories, employs two cathode-ray tube indicators. Within its cabinet is housed the electronic equipment, set up on several chassis which in turn are mounted on racks. This type of unit construction facilitates the insertion of specific circuits for different test applications.

The piece of metal which is to be tested is inserted in a coil which is a part of a tuned circuit of the instrument. This piece of metal constitutes the core of the coil and produces power losses in the tuned circuit which are measurable and are used in various ways to give indication patterns on the cathode-ray tube screens. These core losses are a function of the frequency of the oscillations which are being used. A wide range of frequencies is available for selecting the particular frequency which best differentiates between the standard and the inspected items.

To make sure that the unit is "tamper proof", adjustment controls are mounted on a small panel behind a locked door, the key to which is entrusted only to the plant engineer or supervisor. A shelf rest pulls out of the cabinet, giving the unit the appearance of a metal desk. The proper fixture or coil is in a wooden cabinet with opening or hole to take one by one the items to be inspected. A number of coils are provided where items of different sizes or shapes are to be checked. Below the shelf rest is a double-door cupboard for additional coil fixtures, connecting cables and spare tubes. Indicator lights are employed to show when the instrument is turned on. Receptacles provide for plugging in the coil fixtures and also for external automatic sorting means and for signal lights.

It has been determined that the core losses effected by the metallurgical properties of the metal under test occur for both magnetic and nonmagnetic metals.

(New machines listing on Page 190)



Design Roundup

Broadens Experience With Shock

NOISE ELIMINATION, vibration isolation and shock absorption in machines through use of rubber mountings is not a new development, of course, but applications have expanded tremendously during the war because many types of military materiel have required treatment for these conditions. Experience thus gained will prove valuable in post-war manufacturing of industrial and civilian goods. One of the important developments, born of war necessity, is the discovery that synthetic rubber is in many respects superior to natural rubber for mountings. This, at least, is the conclusion of the General Tire & Rubber Co., Wabash, Ind. Chief advantage of synthetic rubber is that it can be compounded chemically to provide special characteristics required for specific and sometimes exacting applications. Among these characteristics are resistance to fatigue, oxidation, oil, heat and cold.

How Metals Stand Up

EFFECTS ON METALS from a corrosive atmosphere were reported recently to A.S.M.E. by W. L. Maucher and B. W. Jones of the General Electric Co. This report covers four groups of 18 selected metals which were subjected to high temperatures and humidity in the atmosphere of a refinery along the Gulf coast. Chromium alloys in order of merit were 18-8, Nichrome V, 25 per cent chrome steel and 12 per cent chrome steel. While aluminum, untreated, showed a uniform corrosion coating and anodized aluminum had scattered pits, this metal showed up nearly as good as the 18-8. Considering price, anodized or treated-for-paint aluminum would provide one of the most corrosion-resisting combinations.

Copper and its alloys showed that high-copper metals have a low resistance to corrosion in this kind of atmosphere and should not be used unless protected by a coating. Comparison of monel and nickel showed that nickel was not attacked as much as monel. Extremely bad corrosion of cold-rolled steel indicated the necessity for painting or using other protective coatings. Lead showed up outstandingly well but because of its low physicals would not be suitable for many applications.

Although silver was heavily attacked there was no sign of pitting or noticeable etching. Of four electroplated coatings on copper, tin appeared to stand up best. Cadmium was completely gone and silver showed heavy attack. Lead plating did not look bad but inspection disclosed deep pits in the copper base metal.

Facilitates Design Drafting

WHEN DESIGNING units into intricate mechanisms, the designer often finds himself hampered by the complexity of lines representing the units. Quick and simple way to overcome this difficulty is to outline each unit of the mechanism with a different color pencil, working out the additional design on an extra sheet of tracing paper laid over the original layout. This procedure keeps the units clearly defined at all times and, especially in cases where at least two views are utilized, affords the designer a quasi-perspective view not easily obtainable by other means.

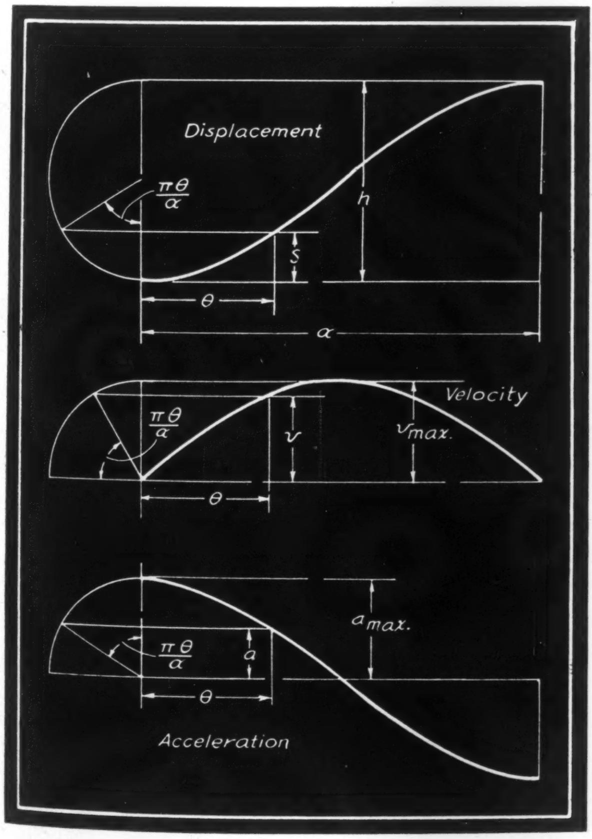
Projection Welding Solves Problem

PROJECTION WELDING methods are being developed to utilize plate steel for sides and base of a small machine instead of conventional methods of fabrication. Although the frame is relatively small it is highly stressed. Arc welding methods were discarded as undesirable because of the difficulty of application and because of the higher cost for quantity production. Experimental models have produced exceptionally high strength values and the appearance of the joints was neat, requiring no finishing operations. In the same machine, tubing has been projection welded to a mechanical part, indicating that the method may be employed in production to effect considerable savings and to develop additional strength as well as reduced weight.

Displacement Diagrams for Cam Design

By Colin Carmichael

FIRST STEP in the construction of a cam profile is the layout of a displacement diagram showing the relation between follower movement and cam rotation. The basic requirement usually is that the follower must be raised a certain distance—the lift—while the cam rotates through a certain angle. Further, a type of motion must be chosen which will not involve abrupt velocity changes. This data sheet presents the methods of constructing displacement diagrams for the types of motion commonly used, together with equations for calculating follower velocities and accelerations


Displacement
$\frac{\pi\theta}{\alpha}$
h
s
θ
α
Velocity
$\frac{\pi\theta}{\alpha}$
v
$v_{max.}$
θ
$a_{max.}$
$\frac{\pi\theta}{\alpha}$
a
θ
Acceleration

Simple Harmonic Motion

For total lift h inches while cam rotates through angle α radians with velocity ω radians per second, the displacement is given by the following equation:

$$s=\frac{h}{2}\left(1-\cos\frac{\pi\theta}{\alpha}\right) \quad \text{(1)}$$

Construction of the displacement diagram is shown at left, the reference semicircle being drawn on a diameter equal to the lift.

Velocity in inches per second is given by:

$$v=\frac{h\pi\omega}{2\alpha}\sin\frac{\pi\theta}{\alpha} \quad \text{(2)}$$

$$v_{max}=\frac{h\pi\omega}{2\alpha}=1.57\frac{h\omega}{\alpha} \quad \text{(3)}$$

Maximum velocity occurs at the middle of the lift.

Acceleration in inches per second per second is

ENGINEERING DATA SHEET

$$a=\frac{h\pi^2\omega^2}{2\alpha^2}\cos\frac{\pi\theta}{\alpha} \quad \ldots\ldots (4)$$

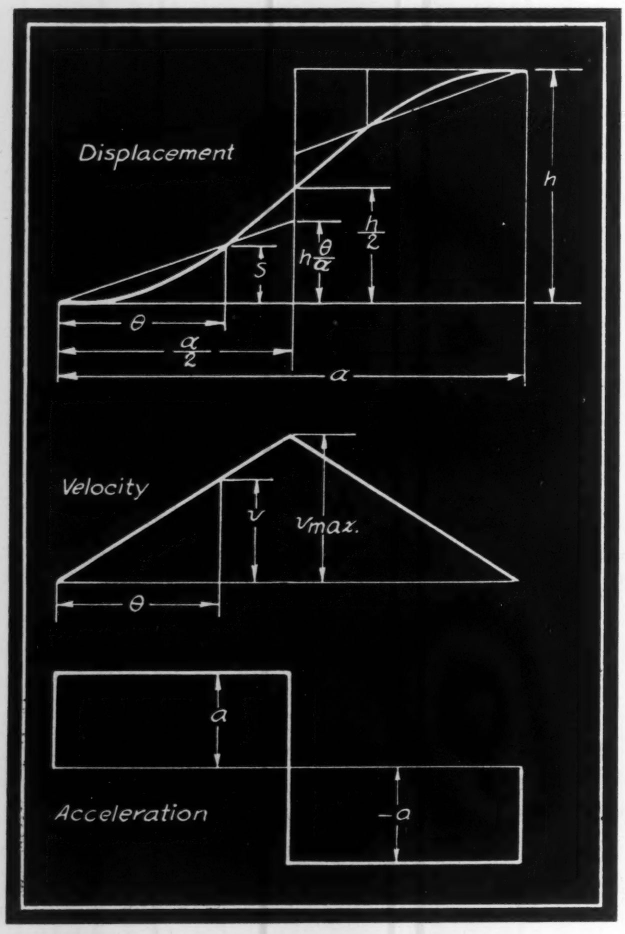
$$a_{max}=\frac{h\pi^2\omega^2}{2\alpha^2}=4.93\frac{h\omega^2}{\alpha^2} \quad \ldots\ldots (5)$$

Maximum acceleration occurs at the beginning of the lift. An equal negative acceleration occurs at the end of the lift.

EXAMPLE: For a lift of 3 inches in one-third of a revolution of the cam, which rotates at 1000 revolutions per minute, it is desired to calculate the maximum velocity and acceleration in simple harmonic motion. Here $\omega = 2\pi \times \text{rpm}/60 = 2\pi \times 1000/60 = 104.7$ radians per second and $\alpha = 2\pi/3 = 2.094$ radians. Therefore, from Equation 3, $v_{max} = 1.57 \times 3 \times 104.7/2.094 = 235.5$ inches per second. From Equation 5, $a_{max} = 4.93 \times 3 \times 104.7^2/2.094^2 = 37{,}000$ inches per second per second or 3080 feet per second per second.

Constant Acceleration and Deceleration

Reduction in the maximum acceleration and hence in the inertia force may be effected through use of constant acceleration and deceleration. Assuming equal values for these two phases, the displacement during acceleration is given by the equation



$$s=\frac{2h\theta^2}{\alpha^2} \quad \ldots\ldots (6)$$

Validity of the construction, which is shown at left, may be checked with this equation. Upper part of the curve (deceleration) is constructed in a similar manner, as shown by the sample construction lines.

Velocity in inches per second is given by

$$v=\frac{4h\omega\theta}{\alpha^2} \quad \ldots\ldots (7)$$

$$v_{max}=2\frac{h\omega}{\alpha} \quad \ldots\ldots (8)$$

Maximum velocity occurs at the middle of the lift, and the velocity diagram is composed of straight lines.

Acceleration in inches per second per second is

$$a=4\frac{h\omega^2}{\alpha^2} \quad \ldots\ldots (9)$$

Acceleration is uniform for the first half of the lift, equal and opposite for the second half.

Comparing Equations 3 and 8 it will be noted that the maximum velocity is 27.4 per cent higher with constant acceleration than with simple harmonic motion. On the other hand, the maximum acceleration (Equations 5 and 9) is 18.9 per cent less, which means a substantial reduction in accelerating force and in the strength of spring required for maintaining contact between follower and cam during the deceleration phase.

ENGINEERING DATA SHEET

Constant Acceleration and Unequal Deceleration

Rapid motion at the start of the lift with a slower deceleration offers some advantages. A lighter spring is adequate to maintain follower and cam contact during deceleration, while in engine valves, for example, wider opening is maintained for a longer time. Assuming the design is based on a predetermined ratio of accelerating to decelerating force (a_1/a_2), the displacement during acceleration is given by

$$s=\left(1+\frac{a_1}{a_2}\right)\frac{h\theta^2}{\alpha^2} \qquad (10)$$

The construction, shown at right, is identical with that for constant and equal acceleration, except that the acceleration phase ends when $\theta=\theta_1$, and $s=s_1$, given by

$$\frac{\theta_1}{\theta_2}=\frac{s_1}{s_2}=\frac{a_2}{a_1}$$

which leads to the relations

$$\theta_1=\frac{\alpha}{1+\dfrac{a_1}{a_2}}$$

$$s_1=\frac{h}{1+\dfrac{a_1}{a_2}}$$

Velocity during acceleration is given by

$$v=2\left(1+\frac{a_1}{a_2}\right)\frac{h\omega\theta}{\alpha^2} \qquad (11)$$

$$v_{max}=2\frac{h\omega}{\alpha} \qquad (12)$$

Maximum velocity occurs at the end of the acceleration phase and the diagram is constructed of straight lines.

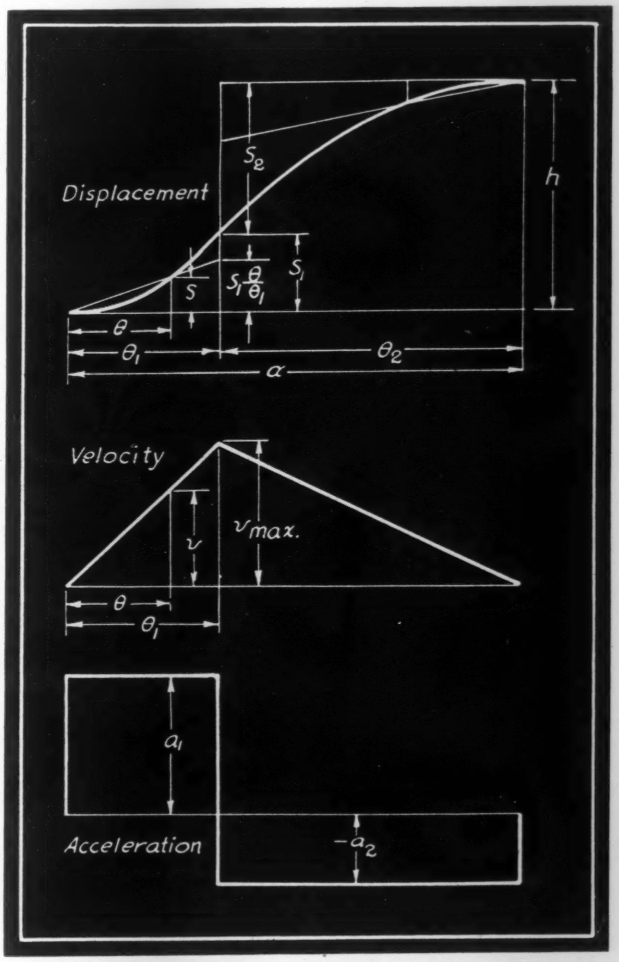
Acceleration is

$$a_1=2\left(1+\frac{a_1}{a_2}\right)\frac{h\omega}{\alpha^2} \qquad (13)$$

Deceleration is

$$a_2=-2\left(1+\frac{a_2}{a_1}\right)\frac{h\omega}{\alpha^2} \qquad (14)$$

(Data Sheet concluded on next page)



ENGINEERING DATA SHEET

Modified Constant Velocity

Because constant velocity involves excessive accelerations at beginning and end of the lift it is necessary, except in exceedingly slow motion, to modify the motion by short acceleration and deceleration phases. While the modifying phases may be simple harmonic or constant acceleration, the latter satisfactorily serves most purposes.

It is assumed that a certain proportion of the lift is to have constant velocity. This ratio is designated s_2/h, leaving s_1/h for acceleration and s_3/h for deceleration as shown in the illustration. In the general case it may be assumed that the accelerations and decelerations are unequal, the ratio being a_1/a_3. The angles θ_1, θ_2 and θ_3 corresponding to the three phases acceleration, constant velocity and deceleration are, respectively,

$$\theta_1=\frac{2\left(1-\frac{s_2}{h}\right)\alpha}{\left(1+\frac{a_1}{a_3}\right)\left(2-\frac{s_2}{h}\right)}$$

$$\theta_2=\frac{\frac{s_2\alpha}{h}}{2-\frac{s_2}{h}}$$

$$\theta_3=\frac{2\left(1-\frac{s_2}{h}\right)\alpha}{\left(1+\frac{a_3}{a_1}\right)\left(2-\frac{s_2}{h}\right)}$$

The displacements for these phases are

$$s_1=h\frac{1-\frac{s_2}{h}}{1+\frac{a_1}{a_3}}$$

$$s_2=h\left(\frac{s_2}{h}\right)$$

$$s_3=h\frac{1-\frac{s_2}{h}}{1+\frac{a_3}{a_1}}$$

From these values the points on the displacement diagram corresponding to the beginning and end of the constant velocity phase may be located. This phase is a straight line, while the other two phases are constructed by the method shown in the illustrations on Pages 146 and 147.

Velocity during the middle phase is

$$v=\frac{h\omega}{\alpha}\left(2-\frac{s_2}{h}\right)\ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ (15)$$

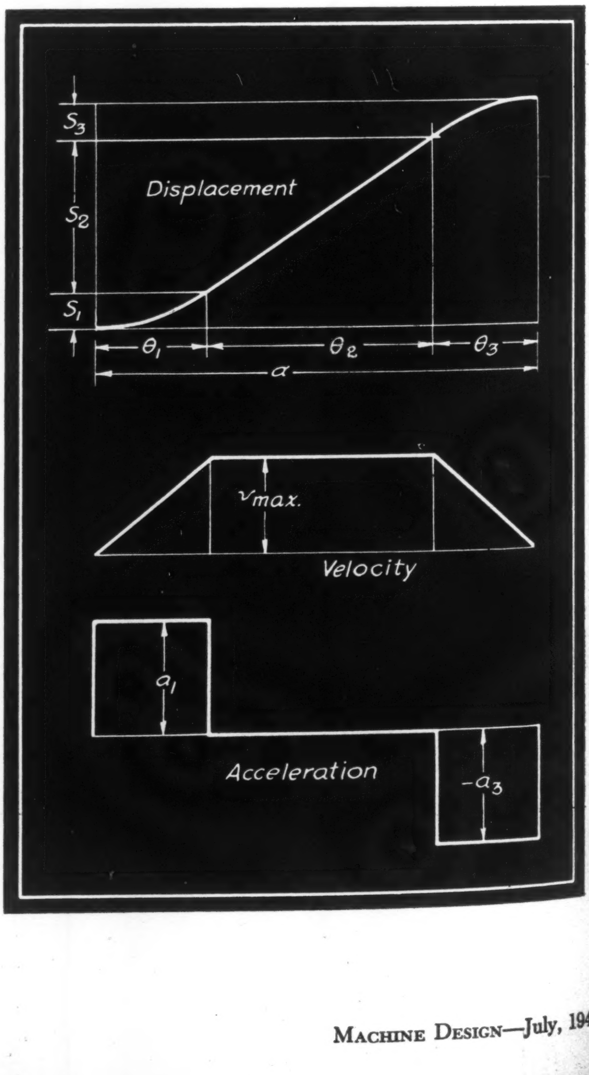
Acceleration is

$$a_1=\frac{h\omega^2}{\alpha^2}\frac{\left(1+\frac{a_1}{a_3}\right)\left(2-\frac{s_2}{h}\right)^2}{2\left(1-\frac{s_2}{h}\right)}\ \ldots\ldots\ldots\ldots\ (16)$$

Deceleration is

$$a_3=\frac{h\omega^2}{\alpha^2}\frac{\left(1+\frac{a_3}{a_1}\right)\left(2-\frac{s_2}{h}\right)^2}{2\left(1-\frac{s_2}{h}\right)}\ \ldots\ldots\ldots\ldots\ (17)$$

Usually the acceleration and deceleration are equal, in which case the foregoing equations are simplified by the substitution of $a_1/a_3=1$.



MACHINE DESIGN'S

Materials Work Sheet

Filing Number 17.01

Aluminum Alloys

2S, 3S and 52S

(wrought, nonheat-treatable types)

AVAILABLE IN:

(2S and 3S)	Sheet, plate, wire, rod and bar, extruded shapes, tubing and pipe, rivets
(52S)	Sheet, plate, wire, rod and bar, tubing and pipe

NOMINAL COMPOSITIONS:

	Manganese	Magnesium	Chromium	Aluminum
(2S)	....	....	....	remainder (99 min)
(3S)	1.2	....	....	remainder
(52S)	....	2.5	.25	remainder

PROPERTIES

YIELD STRENGTH

(.2% offset, typical, psi)

	Alloys		
	2S	3S	52S
0 (soft temper)	5,000	6,000	14,000
¼ hard temper	13,000	15,000	26,000
½ hard temper	14,000	18,000	29,000
¾ hard temper	17,000	21,000	34,000
H (full hard temper)	21,000	25,000	36,000

BRINELL HARDNESS

(500 *kg load,* 10 *mm ball—typical*)

	Alloys		
	2S	3S	52S
0 (soft temper)	23	28	45
¼ hard temper	28	35	62
½ hard temper	32	40	67
¾ hard temper	38	47	74
H (full hard temper)	44	55	85

SHEARING STRENGTH

(typical, psi)

	Alloys		
	2S	3S	52S
0 (soft temper)	9,500	11,000	18,000
¼ hard temper	10,000	12,000	20,000
½ hard temper	11,000	14,000	21,000
¾ hard temper	12,000	15,000	23,000
H (full hard temper)	13,000	16,000	24,000

FATIGUE ENDURANCE LIMIT

(psi)

	Alloys		
	2S	3S	52S
0 (soft temper)	5,000	7,000	17,000
¼ hard temper	6,000	8,000	17,500
½ hard temper	7,000	9,000	18,000
¾ hard temper	8,500	9,500	18,500
H (full hard temper)	8,500	10,000	19,000

THERMAL CONDUCTIVITY

(*at* 100 *deg C—cal/sq cm/sec/deg C/cm*)

	Alloys		
	2S	3S	52S
0 (soft temper)	.54	.45	.37
¼ hard temper	..	.39	..
½ hard temper	..	.38	..
¾ hard temper	..	..	..
H (full hard temper)	.52	.37	.37

MACHINE DESIGN is pleased to acknowledge the collaboration of Aluminum Company of America in this presentation.

TENSILE STRENGTH

(psi)

	Alloys		
	2S	3S	52S
0 (soft temper) typical	13,000	16,000	29,000
max	15,500	19,000	31,000
¼ hard temper, typical	15,000	18,000	34,000
min	14,000	17,000	31,000
½ hard temper, typical	17,000	21,000	37,000
min	16,000	19,500	34,000
¾ hard temper, typical	20,000	25,000	39,000
min	19,000	23,500	37,000
H (full hard temper) typical	24,000	29,000	41,000
min	22,000	27,000	39,000

ELONGATION IN 2 INCHES

(typical, per cent)

	Alloys		
	2S	3S	52S
Sheet Specimen $\frac{1}{16}$-inch thick			
0 (soft temper)	35	30	25
¼ hard temper	12	10	12
½ hard temper	9	8	10
¾ hard temper	6	5	8
H (full hard temper)	5	4	7
Round Specimen ½-inch diam			
0 (soft temper)	45	40	30
¼ hard temper	25	20	18
½ hard temper	20	16	14
¾ hard temper	17	14	10
H (full hard temper)	15	10	8

ELECTRICAL CONDUCTIVITY

(% of Int'l. Annl'd. Copper Std.)

	Alloys		
	2S	3S	52S
0 (soft temper)	59	50	40
¼ hard temper	..	42	..
½ hard temper	..	41	..
¾ hard temper	..	..	..
H (full hard temper)	57	40	40

PHYSICAL CONSTANTS

(all tempers)

	Alloys		
	2S	3S	52S
Specific Gravity	2.71	2.73	2.67
Weight (lb per cu in)	.098	.099	.096
Average Coef. of Thermal Expansion			
68 to 212 deg F	.0000133	.0000133	.000013
68 to 392 deg F	.0000138	.0000138	.0000136

Materials Work Sheet

BEND RADII FOR FLATS, SHEETS AND STRIPS

Since the minimum permissible radii at which these alloys should be bent varies with the nature of the forming operation, type of equipment used and the design and condition of the tools employed, the following recommended minimum radii are intended only as a general guide. The radii are expressed in terms of the stock thickness (t) and apply to 90-degree bends.

	Stock Thickness (inches)					
Alloy	**1/64**	**1/32**	**1/16**	**1/8**	**3/16**	**1/4**
2S-0	0	0	0	0	0	0
2S-1/4H	0	0	0	0	0-1t	0-1t
2S-1/2H	0	0	0	0	0-1t	0-1t
2S-3/4H	0	0	0-1t	½t-1½t	1t-2t	1½t-3t
2S-H	0-1t	½t-1½t	1t-2t	1½t-3t	2t-4t	2t-4t
3S-0	0	0	0	0	0	0
3S-1/4H	0	0	0	0	0-1t	0-1t
3S-1/2H	0	0	0	0	0-1t	½t-1½t
3S-3/4H	0-1t	0-1t	½t-1½t	1t-2t	1½t-3t	2t-4t
3S-H	½t-1½t	1t-2t	1½t-3t	2t-4t	3t-5t	4t-6t
52S-0	0	0	0	0	0	0
52S-1/4H	0	0	0	0-1t	0-1t	½t-1½t
52S-1/2H	0	0	0-1t	½t-1½t	1t-2t	1½t-3t
52S-3/4H	0-1t	½t-1½t	1t-2t	1½t-3t	2t-4t	2t-4t
52S-H	½t-1½t	1t-2t	1½t-3t	2t-4t	3t-5t	4t-6t

APPLICATIONS

2S and 3S: Both are particularly well adapted to parts requiring severe drawing, forming or spinning operations. Their use generally is confined to nonstructural parts such as vats and containers for food products and various chemicals (see Corrosion Resistance), as well as retainers, formed cover plates, shields, filter plates, decorative edging, etc.

52S: Finds general application for moderately stressed parts requiring forming or bending operations which are too severe for the less ductile heat-treated aluminum alloys. For deep drawing, spinning, or extremely severe forming operations where strength is not a primary factor, use alloy 2S. Used for formed housings, spacing rods and tubes, low-pressure hydraulic lines, line fittings, linkage members, etc.

CHARACTERISTICS

All three of these alloys are of the wrought "strain-hardening" type, i.e., their various tempers are produced by subjecting them to definite reductions by cold working after they have been annealed during their fabrication. Sheet, tubing and wire, normally produced by cold working from the hot mill slab or bloom, are available in tempers O (soft), 1/4H, 1/2H, 3/4H, and H, the letter H being an abbreviation for "hard". As will be noted in the tables of properties, the harder the temper the greater the strength of the alloy and the lower the elongation. Bar, rod, plate and shapes (both rolled and extruded) do not receive the precise cold-work processing that sheet, tubing and wire do and consequently are not available in the various tempers. They normally are supplied "as fabricated". Heavy sections of these "as fabricated" forms are of practically the soft temper while thin sections may have properdties approximating those of the half-hard temper.

2S and 3S: Moderate-strength alloys offering lower cost and greater ease of fabrication than the harder alloys. Alloy 2S is commercially pure aluminum and forms more easily than does 3S. However, 3S has greater strength. These two alloys see more use for nonstructural parts than other aluminum alloys and are available in the widest range of sizes.

PROPERTIES AT ELEVATED TEMPERATURES

YIELD STRENGTH

(*typical, psi*)

	Temperature (degrees Fahr.)					
	75	300	400	500	600	700
2S-O (soft temper)	5,000	3,500	3,000	2,000	1,500	1,000
2S-1/2 hard temper	14,000	10,000	6,500	2,000	1,500	1,000
2S-H (full hard temper)	21,000	14,000	3,000	2,000	1,500	1,000
3S-O (soft temper)	6,000	5,000	4,500	3,500	2,500	2,000
3S-1/2 hard temper	18,000	15,000	9,000	5,000	3,000	2,000
3S-H (full hard temper)	25,000	16,000	8,000	5,000	3,000	2,000
52S-O (soft temper)	14,000	13,500	11,000	8,000	4,000	2,500
52S-3/4 hard temper	34,000	27,000	11,000	8,000	4,500	2,500

TENSILE STRENGTH

(*typical, psi*)

	75	300	400	500	600	700
2S-O (soft temper)	13,000	7,500	6,000	3,500	2,500	1,500
2S-1/2 hard temper	17,000	13,000	9,500	3,500	2,500	1,500
2S-H (full hard temper)	24,000	17,500	6,000	3,500	2,500	1,500
3S-O (soft temper)	16,000	11,000	8,000	5,500	4,000	3,000
3S-1/2 hard temper	21,000	18,000	14,000	10,500	6,000	3,000
3S-H (full hard temper)	29,000	23,000	17,000	10,500	4,500	3,000
52S-O (soft temper)	29,000	23,000	18,000	12,000	7,500	5,000
52S-3/4 hard temper	39,000	32,000	25,000	12,000	8,000	5,000

ELONGATION IN 2 INCHES

(*typical, per cent*)

	75	300	400	500	600	700
2S-O (soft temper)	45	65	70	85	90	95
2S-1/2 hard temper	20	22	25	85	90	95
2S-H (full hard temper)	15	16	70	85	90	95
3S-O (soft temper)	40	47	50	60	60	60
3S-1/2 hard temper	16	17	22	25	40	60
3S-H (full hard temper)	10	12	15	25	55	60
52S-O (soft temper)	30	55	65	100	105	120
52S-3/4 hard temper	10	16	35	80	100	120

52S: Has twice the strength of 2S-O in the annealed temper. Cold working strain-hardens it more rapidly than 2S, thus the ease and cost of its fabrication is not as favorable. Has mechanical properties intermediate between those of 2S and 3S on the one hand and the high-strength, heat-treatable alloys on the other; is intermediate in cost also. Endurance limit of this alloy is higher than all other wrought aluminums.

Various tempers of alloys 2S, 3S and 52S cover a strength range of from 12,000 to 40,000 pounds per square inch. Where higher strength is required of a wrought aluminum alloy, one of the heat-treated alloys must be used. All three of these alloys have high resistance to corrosion.

FABRICATION

All three of these alloys are machined readily. Compared to the tools used for cutting steel, tools for machining aluminum have appreciably more side and top rake, their shapes, in fact, closely approximating tools used on hard woods. By finishing on a fine abrasive wheel followed by handstoning or lapping, the tool edges are kept sharp and smooth. In lathe cutting, the tool is set appreciably higher on the work than with steel. For drilling, twist drills having large spiral angles generally are preferred.

Best results usually are obtained by use of relatively high speeds in conjunction with fine to medium feeds; the finer the feed, the higher the speed. Lubricant is used copiously. As a cutting compound for general use, a mixture of kerosene and lard oil is satisfactory, while for milling, sawing and drilling, the less expensive soluble cutting oils are used.

DRAWING:

The three alloys covered by this work sheet are well suited to the production of drawn shapes. The temper employed will depend on the severity of the draw, annealed sheet being used for real deep draws while half-hard or even full-hard sheet may be satisfactory for less severe draws. As successive draws are effected, the metal strain-hardens and changes from the annealed to the harder tempers with corresponding increases in mechanical properties as shown in the table "Effects of Drawing on Mechanical Properties".

In the following table are listed the recommended reductions in diameter per draw for deep-drawn cylindrical shells.

Reduction in Diameter for Deep Shells
(*alloys* 2S-0 *and* 3S-0)

Operation		Desired Reduction (per cent)	Permissible Reduction (per cent)
Blank	(D)		
First Draw	(D_1)	40 (or less) D	42D
Second Draw	(D_2)	$20D_1$	$25D_1$
Third Draw	(D_3)	$15D_2$	$18D_2$
Fourth Draw	(D_4)	$15D_3$	$15D_3$

Little change in hardness is effected after the third draw. Thus, the 15-per cent value given in the immediately preceding table is suitable for the fourth and all subsequent draws. Minimum radius for the die is four times the original sheet thickness; maximum radius is fifteen times. Radius on punch should be not less than four times the thickness. Too sharp a radius will cause excessive resistance to flow of the metal and result in fractures, while too large a radius is apt to cause wrinkling of the side walls of the shell. Deep drawing requires thorough lubrication. Most often used are mineral oils or compounded mineral oils.

BLANKING AND PIERCING:

Usually the punch is made of annealed tool steel and the die is made of hardened tool steel. Clearances between punch and die, less for the soft than for the hard tempers, are listed below. Use clearances of about one-half the value of those listed for new punch and die sets, as wear, especially on the punch, will increase the clearance as the tools are used.

Punch and Die Clearances for Blanking
(t=thickness of sheet in inches)

Sheet Alloy No.	Clearance: On a Side	Clearance: On the Diameter
2S-0	.10t	.20t
2S-1/2 H	.12t	.24t
2S-H	.14t	.28t
3S-0	.10t	.20t
3S-1/2 H	.12t	.24t
3S-H	.14t	.28t
52S-0	.13t	.26t
52S-1/2 H	.14t	.28t
52S-H	.15t	.30t

The following table shows the recommended scrap on the sides and between blanks for various thicknesses of sheet:

Scrap Allowances for Blanking

Thickness (inches)	Blank Diam. (inches)	Scrap on the side (inches)	Scrap between blanks (inches)
.013 to .057	0 to 10 incl.	1/8	1/16
	10 to 20 incl.	3/16	3/32
	20 to 30 and over	1/4	1/8
.057 to .081	0 to 10 incl.	5/32	3/32
	10 to 20 incl.	7/32	1/8
	20 to 30 and over	5/16	5/32
.081 to .128	0 to 10 incl.	3/16	1/8
	10 to 20 incl.	1/4	5/32
	20 to 30 and over	3/8	3/16

Effects of Drawing on Mechanical Properties

Alloy No.	No. of draws	Thickness (inches)	Tensile Strength (psi)	Increase (per cent)	Yield Strength (psi)	Elongation in 2 inches (per cent)	Approx. Hardness
3S	0	.103	16,000	....	6,000	30	0
	1	.100	18,740	17.2	16,700	11	1/4H
	2	.097	22,140	38.4	20,800	9	1/2-3/4H
	3	.097	23,710	48.2	21,900	8	3/4H
	4	.098	24,240	51.5	22,300	7.5	3/4H
52S	0	.103	29,150	....	14,200	27	0
	1	.101	34,390	18.0	31,600	6	1/4-1/2H
	2	.090	39,710	36.2	37,100	5	3/4H
	3	.087	42,680	46.5	38,600	5,5	H
	4	.089	43,750	50.2	36,100	6	H

Materials Work Sheet

BLANKING AND PIERCING: *(cont'd.)*

When blanking hard aluminum sheet, both the tool cutting edges and the sheet are lubricated with a medium grade of engine oil plus a small percentage of fatty oil diluted with kerosene. For sheets of the softer alloys or tempers, the engine oil and fatty oil mixture is used without dilution.

RIVETING:

For riveting aluminum sheet and plate, rivets of aluminum alloys 2S, A17S and 53S usually are employed. In general, the diameter of rivets should not exceed two and one-half to three times the thickness of the sheet or plate, nor should their diameter be less than the thickness of the thickest plate through which they are driven. Recommended minimum spacing is three times the nominal rivet diameter and maximum spacing should not exceed twenty-four times the thickness of the sheet or plate. For maximum bearing strength, the edge distance measured from the center of the hole in the direction of stressing should be at least twice the diameter of the rivet hole.

When riveted joints must be pressure-tight, they may be caulked either by deforming the edges of the plates with a caulking tool or by using caulking strips of a softer alloy. Joints which are to be caulked should have edge distances about one and one-half times the rivet diameter or about four times the thickness of the plate. Spacing of rivets on a caulked joint should not exceed four times the rivet diameter or about ten times the thickness of the plate. The joint may be packed with a sealing material which is nonabsorbent, firm, and tough enough not to crack or spall. Such packing materials should be noncorrosive to aluminum and should be applied in layers thin enough not to affect the strength of the joint appreciably.

Rivet holes may be punched, drilled, or sub-punched and reamed to size, the last mentioned method being preferable, particularly when reaming is done in assembly to insure exact coincidence of the holes. The best clearance between rivet and hole is the smallest one which will allow the rivet to be inserted easily without delay. In general, the rivet should have about the same properties as the alloy into which it is driven. The following table lists the bearing strengths of aluminum alloy plates and shapes representative of material having typical properties:

Bearing Strengths

Alloy No.	Yield Strength (psi)	Ultimate Strength* (psi)
2S-0	13,000	23,000
2S-1/4 H	15,000	27,000
2S-1/2 H	17,000	30,000
2S-3/4 H	20,000	36,000
2S-H	24,000	43,000
3S-0	16,000	28,000
3S-1/4 H	18,000	32,000
3S-1/2 H	21,000	38,000
3S-3/4 H	25,000	45,000
3S-H	29,000	52,000
52S-0	29,000	52,000
52S-1/4 H	34,000	61,000
52S-1/2 H	37,000	67,000
52S-3/4 H	39,000	70,000
52S-H	41,000	74,000

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*These values should be used only when the edge distance measured from the center of the rivet hole in the direction of stressing is not less than twice the diameter of the rivet hole. For smaller edge distances values should be reduced proportionately.

SPINNING:

Alloy 2S lends itself more readily to spinning than do any of the other aluminum alloys. Alloy 3S often is used because it has greater strength than 2S, but it may require intermediate annealing on deep spinning as well as more breakdown operations. Although 52S occasionally is spun, it requires more frequent reannealing that 3S because of its more rapid rate of strain hardening. Area of the blank should be equal to the surface area of the finished part. Most practical lubricant, especially for large spinnings, is ordinary laundry soap. It is applied to the blanks before and during spinning.

GAS WELDING:

Commonly used gas processes employ oxyhydrogen or oxy-acetylene flames. Both of these give equally good results. Proper choice of filler material is important. Alloy 2S generally is gas welded with metal of the same purity as the parent metal. For 3S, filler rod of 2S is in most cases considered best, and for 52S, filler rod containing approximately 95 per cent aluminum and 5 per cent silicon (43S) generally is used. Sheet which is ⅜-inch and greater in thickness usually is preheated to 700-800 degrees Fahr., care being taken not to exceed the upper limit. Edge or corner welds give rise to the least amount of distortion. Minimum distortion is obtained also in assemblies where the weld bead is located on a crown or radius. Welded seams and fittings in flat areas are to be avoided because of warping and buckling. Flux is required to break down the oxide coating which is present on the surface of all aluminum and aluminum alloys. To clean off flux residual deposits after welding, boiling water and a fiber brush may be used. Where this method is not feasible, the part may be immersed in 10 per cent sulphuric acid for 30 minutes or in cold 10 to 50 per cent nitric acid for 10 to 20 minutes. In any case, where acid is used, the dip must be followed by thorough rinsing in hot or cold water.

METAL-ARC WELDING:

Offers the advantage of a concentrated heated zone which prevents excessive expansion of the parts and lessens distortion. However, the arc is controlled with relative difficulty, the result being that it is not practicable to butt weld joints on sheet and plate lighter than ⅛-inch thick. Flux-coated electrodes must be used. No edge preparation is necessary on material ¼-inch or less thick.

CARBON-ARC WELDING:

Only automatic arc-welding method used for joining aluminum alloys on a mass production basis. No edge preparation necessary but backing up weld is essential. Same filler rods used as are employed for gas welding. Flux residue is removed in same manner also.

ATOMIC-HYDROGEN WELDING:

Another arc-welding process offering good results. It, like carbon-arc welding, can be effected manually or automatically. Butt welds can be made with material 1/16-inch thick and heavier. Preparation of joints and removal of flux residue are accomplished by same methods as are employed with gas-welded joints.

SPOT WELDING:

Parts must be so designed as to permit their being placed in the welding machine with the work surfaces normal to the axis of the electrodes and with no other portion of the part in contact with any uninsulated portion of the welding machine. (Throat depth of average spot welder is about fifty inches.) Also, because of the size and shape of the electrode, the minimum distance from the center of the weld to flanges one inch high or less is 5/16-inch and is ½-inch where the flange height is over one inch. For best results distance from center of weld to edge of sheet should be at least four times the sheet thickness and the distance between spots should be at least eight times the sheet thickness. Spot welds in these three alloys usually are as resistant to corrosive attack as are the adjacent unwelded areas. Parts should be cleaned thoroughly before spot welding is attempted. These alloys may be seam or flash-welded also.

BRAZING:

While alloys 2S and 3S can be brazed satisfactorily on a commercial basis, alloy 52S is more difficult to handle and is not suitable for all jobs. Since brazing requires heating these alloys above their annealing temperatures, designs must be based on the strength of the soft temper of the material. Three methods of brazing are employed: Furnace brazing, torch brazing and dip brazing. Generally, the cost of brazing is less than that of either gas or arc welding and brazed joints are neater and require less finishing. In addition, parts too thin to be welded may be brazed satisfactorily. Lap joints usually are employed rather than butt or scarf joints and clearance always is provided between the parts for flow of the filler material. For laps less than ¼-inch long, clearances of .006 to .010-inch are suitable. Longer laps require clearances up to .025-inch.

CORROSION RESISTANCE

Alloy 2S offers high general resistance to corrosion. Alloy 3S has practically the same resistance as 2S to the atmosphere and to salt water. Alloy 52S appears to be more resistant to salt water than 2S from the standpoint of the retention of both its mechanical properties and its appearance. Anodic coatings, which can be applied in various colors (by die or pigment adsorption) provide good resistance to corrosion, have high dielectric strength and can be made hard so as to offer resistance to abrasion and durability to the part. For maximum resistance to corrosion, anodic coatings should be sealed with a chromate or other sealer. Electroplated chromium finishes sometimes are applied where corrosion and abrasion resistance is required.

GALVANIC CORROSION

Contact of these alloys with copper, brass, nickel, bronze, monel, iron and steel should be avoided in the presence of salt water or sea coast atmospheres, as these metals are cathodic to aluminum and the resulting galvanic couple results in accelerated corrosion of the aluminum. Zinc and cadmium-plated parts, however, can be used safely in contact with them.

ANNEALING

Strain hardening, resulting from cold working, may be removed by annealing. Complete softening is practically instantaneous for alloys 2S and 52S at temperatures greater than about 650 degrees Fahr., and for 3S at temperatures of 750 degrees Fahr. or higher. The same results may be accomplished by heating for longer periods of time at somewhat lower temperatures. If the metal has reached its instantaneous annealing temperature, the exact temperature is not critical, although the recommended values should not be exceeded by too much nor held at higher temperatures for excessive lengths of time. While rate of cooling is not important, rapid cooling may impair the flatness of the material.

Materials Work Sheet

MATERIAL DESIGNATIONS

Army-Navy Aeronautical (AN)

(2S) AN-FF-R-551 (rivets)
(2S) AN-QQ-W-298 (rivet wire and rod)

Federal

(2S) QQ-A-561 (sheet and plate)
(2S) QQ-A-411 (wire, bar, rod and shapes)
(2S) WW-T-783 (round tubing)
(2S) QQ-R-571 (welding rods)
(3S) QQ-A-359 (sheet and plate)
(3S) QQ-A-356 (wire, bar, rod and shapes)
(3S) WW-T-788 (round tubing)
(52S) QQ-A-318 (sheet and plate)
(52S) WW-T-787 (tubing)

Army Air Forces

Same as Federal.

Navy

(2S) 47A2 (sheet and plate)
(2S) 46A3 (wire, bar, rod and shapes)
(2S) 44T19 (round tubing)
(2S) 43R5 (rivets, rivet wire and rod)
(2S) 46R7 (welding rods)
(3S) 47A4 (sheet and plate)
(3S) 46A6 (wire, bar, rod and shapes)
(3S) 44T20 (round tubing)
(3S) 43R5 (rivets, rivet wire and rod)
(52S) 47A11 (sheet and plate)
(52S) 44T32 (tubing)
(52S) 46A11 (wire, bar and rod)

DATA ON STOCK FORMS

Commercial Thickness Tolerances

(plus or minus)

FLAT SHEET, ALLOYS 2S and 3S

Tolerance in Per Cent of Nominal Thickness or in Inches

Standard Thickness (inches)	Widths up to 36″	Widths over 36 to 54″	Widths over 54 to 72″	Widths over 72 to 90″	Widths over 90 to 102″
.010	.0015	....	...	...	...
.012, .014, .016	.0015	.002	...	...	...
.018, .020 .025, .032	.002	.0025	...	...	...
.040	.0025	.003	.004	...	...
.051, .064	.003	.004	.005	.007	...
.081, .091	.003	.004	.006	.008	...
.102, .125	.0045	.005	.007	.009	.010
.156, .188 .204, .249	4%	5%	6%	7%	8%

FLAT SHEET, ALLOY 52S

Tolerance in Per Cent of Nominal Thickness or in Inches

Standard Thickness (inches)	Widths up to 36″	Widths over 36 to 42″	Widths over 42 to 48″	Widths over 48 to 54″	Widths over 54 to 60″	Widths over 60 to 66″	Widths over 66 to 72″
.010, .012 .014 *	.0015	...	...	...	...	...	...
.016, .018	.002	...	...	...	...	...	...
.020	.002	.003	...	...	...	...	...
.025	.0025	.003	...	...	...	...	...
.032	.0025	.003	.004	...	...	...	...
.040	.0025	.003	.004	.004	.005	...	...
.051, .064	.003	.004	.004	.005	.006	...	...
.081, .091	.003	.004	.004	.005	.006	.008	.010
.102, .125	.0045	.005	.005	.005	.007	.010	.012
.156, 188 .204, .249	4%	5%	5%	5%	6%	8%	10%

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*Tolerance applies up to maximum width of 30 inches in these thicknesses.

PLATE, ALLOYS 2S, 3S and 52S

Tolerance in Per Cent of Nominal Thickness

Thickness (inches)	Widths up to 54″	Widths over 54″ to 72″	Widths over 72″ to 90″	Widths over 90″ to 120″
.250 to .374	5	6	7	8
.375 to .500	5	5	6	7
.501 to 1.000	4	4	5	6
1.001 to 3.000	3	3	4	5

Commercial Tolerances for Wire, Rod and Bar

ROLLED ROUND ROD

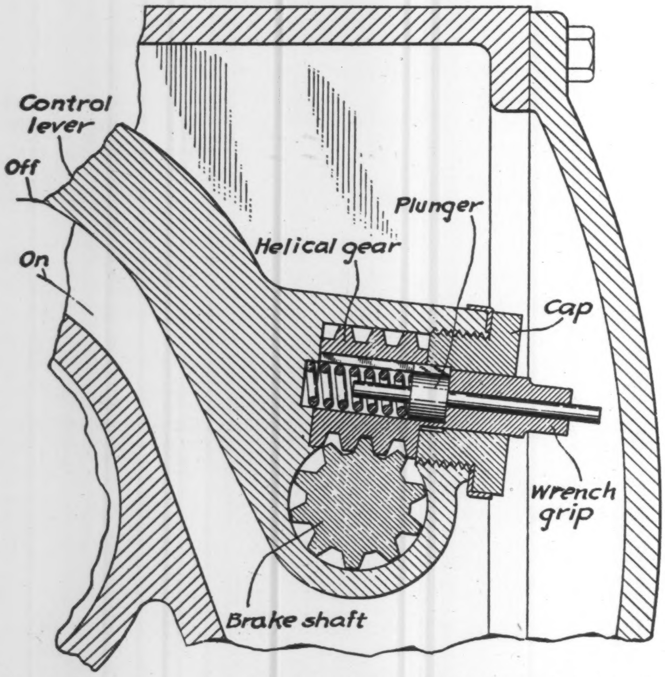
Diameter (inches)	Tolerance (inches) Plus	Tolerance (inches) Minus
1.501 to 3.499	.008	.008
3.500 to 5.000	$\frac{1}{32}$	$\frac{1}{64}$
5.001 to 8.000	$\frac{1}{16}$	$\frac{1}{32}$

Noteworthy Patents

Takes Up Slack From Brake Wear

SIMPLIFIED means for taking up the slack incident to wear of a brake are covered by patent 2,337,756 recently assigned to the Westinghouse Air Brake Co. The adjustment is simple to make and the device incorporates positive locking against slip.

Accompanying illustration shows the design applied to the connection between a control shaft which operates the brake and the lever by which it is actuated. This lever extends at right angles to the axis of the shaft and has one end journaled on it. Within the journaled end the shaft is provided around its periphery with gear teeth meshing with helical teeth provided on a gear whose axis is at right angles to the brake shaft. A cap secured to the end of the brake lever closes the end of the bore in which the helical gear is disposed, holding it against axial movement. Engagement between the teeth on the brake shaft and gear provides a rigid drive connection between these parts, so that when the lever is rotated counterclockwise the shaft will turn in the corresponding direction to apply the brake.



Control lever is connected to brake shaft through helical gears, rotation of which changes relative position to compensate for brake wear

Because the zone of movement indicated by the off and on position on the diagram must be substantially maintained throughout the life of the brake, periodical change in the angular position of the brake shaft with respect to the lever to compensate for wear is necessary. This angular changing of the position is effected by rotation of the helical gear which, in turn, rotates the brake shaft within the end of the lever.

For turning the helical gear an extension is provided which passes through a bore in the shaft and ends in a square, hexagon or other shape suitable for gripping with a wrench. A plunger which can slide in a bore within the helical gear is used for locking purposes. This plunger is provided with a key or tongue, projecting through a slot in the extension of the gear. This tongue may engage any one of several angularly spaced slots in the shaft against which the plunger is pressed by a compression spring.

In order to make an adjustment it is necessary first to press the end of the stem, moving the plunger against the spring and releasing the tongue from the slot in the shaft. The gear may then be turned to take up slack following which the force on the pin may be removed to permit the spring to force the plunger back to its normal position in which the tongue locks the gear to the shaft, holding the brake shaft in the adjusted condition. If desired the wrench may be of such a type as to move the plunger in the act of applying the wrench, thus greatly facilitating the adjustment.

Provides Speed Reduction in Wheel Hub

AXLE shafts of heavy trucks often are subjected to severe torsional stress, especially when starting the vehicle in soft ground, with the result that breakages are not infrequent. By incorporating part of the overall speed reduction in the hubs, so that the axle shafts rotate at a higher speed than the wheels, the torsion in these shafts is reduced. A design employing spur gearing between each axle shaft and the respective driving wheel hub is covered by patent 2,337,884, recently assigned to The Austin Motor Co. Ltd.

In its simplest form the reducer consists of a spur wheel fixed on the outer end of the axle shaft and gearing with one or more intermediate pinions which rotate about axes fixed in relation to the axle housing. These intermediate pinions also engage with an internally toothed ring gear attached to the wheel hub. With this arrangement the speed ratio is represented by the ratio of teeth on the axle shaft spur gear to the teeth on the internal ring gear. Alternative arrangements include the employment of compound pinions, permitting more latitude in providing several ratios within the same overall dimensions; and the use of idler pinions to make the direction of rotation of the wheel the same as that of the axle shaft.

If three or more pinions are employed to divide the load, the support which they afford the spur gear makes it unnecessary to provide a separate bearing at the outer end of the axle shaft.

PROFESSIONAL VIEWPOINTS

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

". . . on cemented metal assemblies"

To the Editor:

Analyses of the characteristics and limitations of current adhesives as set forth in John Delmonte's article in the May issue, "Cemented Assemblies—Their Place in Design", are in accord with our developments and opinions concerning cemented assemblies. The many advantages of cemented assemblies are quite apparent.

We feel that this type of assembly has its place in machine construction, particularly where dissimilar materials are joined; for example: Metal and wood, or metal and rubber, or any such combinations.

It is well to keep in mind, however, that (1) in cementing various metal assemblies the surface preparation (or cleaning) is exacting, and (2) that it must be certain that the areas to be bonded are intimately in contact with each other by utilization of great pressures on the joint while the cement is being cured. Therefore, elaborate dies and costly press equipment are required. As expressed in the article, this is true when solvent-carrying adhesives are used.

In aircraft construction the constant necessity for design changes along with a limited production as might be expected in the postwar period can change the industry's viewpoint on cemented metal assemblies, this being due to initial outlay for production equipment. The advent of a new adhesive which has the remarkable characteristics of cycleweld, yet is nonsolvent carrying, would eliminate the requirement of high pressures in making a cemented joint. Then cementing metal assemblies would be practical even in small quantities.

—H. R. LEADER
Material and Process Engr
Bell Aircraft Corp.

". . . drives requiring constant power"

To the Editor:

Your April MACHINE DESIGN is an excellent number, and the article "Specifying Variable-Speed Drives" in this issue is especially interesting. We note, however, a relatively minor point which is not fully correct. A conveyor drive is customarily thought of as a constant torque drive. The three most common drives requiring approximately constant power over the speed range are machine tool spindles, mixers and winders. In the first two of these, the torque goes up with reduced speed, since the speed usually is lowered to overcome a greater resistance. In the last case, where it is desired to wind at constant linear speed, the revolutions per minute of the drum, of course, must be reduced (usually automatically) as the material winds up to a bigger diameter, and as the tension usually remains constant, the torque obviously increases in proportion to the increased diameter. In fact, the automatic control systems used on jobs of this kind, such as paper winding, often are based on the driving motor requiring constant power during the winding operations.

—L. A. GRAHAM
Graham Transmission

". . . methods of changing speed"

To the Editor:

The article, "Specifying Variable-Speed Drives, Part II —Electrical", in the May issue, covers quite thoroughly the available methods of changing speeds electrically. The two characteristics often misinterpreted and misused by machine designers are emphasized. These are: (1) Adjustable versus variable speed, and (2) constant torque versus constant horsepower. "Variable speed" has become the all-inclusive term as evidenced by the title of this article, instead of the more correct "changeable speed". It is important that the constant torque characteristic of Ward-Leonard armature control of direct-current motors, either through a motor-generator set or electronic means, be noted.

Present trend to high-speed motors has again brought out two facts: (1) The simplicity and ruggedness of the squirrel-cage induction motor, and (2) the need for a reliable, compact frequency changer with stepless control, preferably electronic. The trend in grinding machines is to mount the wheel motor directly on the wheel spindle, instead of using a belt connection, for compactness and saving in power. Two requirements for maintaining correct peripheral speed are evident: Small wheels require high speeds, and different size wheels and wheel wear compensation require adjustable speed.

The direct-current motor using Ward-Leonard control is limited to relatively low speeds, requires some maintenance, and is difficult to build into a spindle. The squirrel-cage induction motor with frequency control is ideal. The frequency converter is well known to the woodworking industry. This consists of a wound-rotor motor acting as a generator and driven by any adjustable speed means, such as a multispeed motor, or a constant-speed motor with adjustable-pitch pulleys, or stepped pulleys with belt shifting where changes are infrequent. The frequency range, however, is not proportional to the speed range of the driving means.

An electronic frequency changer would appear to have advantages over the widely advertised electronic voltage control for direct-current motors.

—L. MICHELSON, *Electrical Engineer*
Gleason Works

". . . number of rivets in aircraft"

To the Editor:

As a matter of interest in connection with the reference to the use of about 3,000,000 rivets throughout the structure of a present-day large-sized craft, which appeared in the section, "This Issue at a Glance" of the March issue, the actual count of the number of rivets and the approximate gross weight in pounds of three airplanes are as follows:

Airplane	Rivets	Gross Weight
Cargo	450,000	45,000
Fighter	85,000	9,000
Fighter	150,000	12,000

Based on the above figures, the number of rivets per pound gross weight for these airplanes are 10, 9.5 and 12.5 respectively. These three airplanes are land types and it would not seem unreasonable to use a figure of 15 rivets per pound gross weight for seaplane types.

Considering airplanes of 150,000 pounds gross weight (Martin Mars, for example), the above reasoning leads to a figure of 2,250,000 rivets in its structure. Since airplanes of much larger gross weights than the Mars are now under development, the figure of 3,000,000 rivets does not seem to be out of line.

—MARTIN GOLAND
Research Laboratory
Airplane Division
Curtiss-Wright Corp.

". . . close tolerances by hardening"

To the Editor:

I have noted in the "Materials Work Sheet" dealing with beryllium copper which appeared in the May issue of MACHINE DESIGN that an annealing temperature of 800 degrees Cent. is recommended. It has been my experience that the most satisfactory results are obtained by annealing this material at 775 degrees Cent.

In the section of the work sheet dealing with forming, it might have been well to mention the property of beryllium copper whereby it can be made to conform to extremely close tolerances by hardening in fixtures. The metal becomes quite plastic during the hardening operation and will accommodate itself to the shape of the fixture if the part and the fixture have been made properly. In many cases closer tolerances can be held by this means than can be obtained from cold forming with allowance for spring-back.

—W. E. MARTIN
Materials Laboratory
Sperry Gyroscope Co.

". . . formulas for rosette analysis"

To the Editor:

With reference to my article "Graphical Method Simplifies Strain-Rosette Analysis," which appeared in the May issue of MACHINE DESIGN, your readers may be interested in the official formulas for rosette analysis with Baldwin-Southwark SR-4 gages, recently developed by Professor Ruge of Massachusetts Institute of Technology. The analysis is carried out somewhat differently from the one shown in my article. However, the two methods can be correlated and the accompanying table shows the conversion of coefficients.

Conversion Table

SR-4 Rosette Configuration and Type Number	Baldwin-Southwark Factors*	MACHINE DESIGN Factors*
Rectangular Rosettes (Types AR-1 and CR-1) T-Δ Rosettes (Type AR-3)	a	$F(1-K)$
	b	$\frac{1}{K}$
	$\frac{ab}{b-1}$	F
	$\frac{1}{b}$	K
Equiangular Rosettes (Types AR-4 and CR-4)	a	$\frac{3F(1-K)}{3+K}$
	b	$\frac{3+K}{2K}$
	$\frac{ab}{b\text{-}2}$	F
	$\frac{3}{2b-1}$	K

---

*Key for Factors:
a = Modified gage sensitivity.
b = Compensation factor for transverse gage sensitivity.
F = Strain sensitivity of filament (physical constant).
K = Specific transverse sensitivity of strain gages (geometrical factor).

Incidentally, in my article the caption for *Fig.* 1 should refer to a "240-millimeter howitzer" instead of "24-millimeter", and the illustrations for *Figs.* 2 and 3 are interchanged.

—J. H. MEIER, *Research Engr.*
Bucyrus-Erie Co.

We apologize for the errors and hope they did not lead to any confusion on the part of our readers.—ED.

Soundness of castings and forgings may be tested by a hammer blow in conjunction with a microphone pickup. The sound picked up is amplified and filtered to suppress harmonic frequencies. Remaining base frequency determines whether the part being tested is sound or defective.

Torque Converter

(*Continued from Page* 122)

more power at stalling through racing, and thus the engine speed at stalling decreased to about 70 per cent of racing speed. This is important for automotive drives as it prevents over-speeding of the engine at stalling and flattens out the efficiency curve.

However, an increased stalling torque of about 15 to 20 per cent can be obtained by using long blades in the last turbine stage, reducing the ratio mentioned to 80 per cent. This is useful for most industrial applications where engine speeds generally are lower. A further increase in stalling torque of 15 per cent is obtainable by using adjustable pump blades which also permit control of the input power.

Range of Torque Converter: In order to obtain the correct performance from a torque converter, it must be used with an engine of suitable torque-speed characteristic in the same way that a ship's propeller must be designed to suit the engine characteristic. Certain variations from the standard characteristic can, however, be made without material alteration of design or efficiency. The easiest ways to alter the torque-speed characteristic are as follows:

1. To reduce the external diameter of the pump wheel if it is desired to use an engine of lower torque
2. To increase the effective outlet angle of the pump blades by increasing the number of blades or the blade angle for an engine of higher torque
3. To use third turbine blades, which increases the input power considerably at high-speed ratios
4. To incorporate a primary step-up or reduction gearing.

Some experiments showing the effect of altering the blade angle are illustrated in *Fig.* 11. The efficiency curves are shown for constant primary torques, and the decreased range of the converter and lower efficiency as the angle is decreased will be noted. The available primary-speed range also is shown on the curves for each angle. The curves show the great importance of having pump blades which have been determined carefully and are accurate.

The effect of reducing the pump wheel diameter is shown in the curves in *Figs.* 12 and 13. It will be noted that as the diameter is reduced the efficiency at the lower end of the curve is reduced, and is increased at the higher end. Reducing the diameter displaces the curves along the secondary speed axis owing to the increased input. *Fig.* 13 shows the alteration of the stalling characteristic produced by reducing the diameter of the pump wheel by various amounts.

From these results a suitable pump wheel diameter to suit any particular engine characteristic or condition can be determined. The efficiency curves in *Fig.* 12 are shown for one constant primary torque only and represent only one condition. It will be noted in *Fig.* 13, that the torque ratio at stall increases with decrease of wheel diameter in approximately inverse proportion to the diameter.

Torque Converter With Adjustable Pump Blades: In *Fig.* 14, characteristic curves are given for a torque converter having adjustable pump blades. The numbers of the curves indicate different positions of the adjustable blades.

It will be noted that the unit's efficiency is less sensitive for different blade positions than in the case of the pump blades corresponding to *Fig.* 11. Advantages of the adjustable blades are:

1. Practically no secondary torque with closed blades
2. Higher stalling torque ratio, up to 6:1
3. Flatter efficiency curve
4. Possibility of overloading the engine when desired
5. Possibility of avoiding critical speeds of the engine crankshaft.

Effect of Turbine Variations: It is a simple matter to alter a pump wheel on a production converter, as it may have only from 14 to 20 blades, but it is not so easy to make turbine-blade alterations, because of the large number of blades. However, as already mentioned, two different types of third-stage blades are used, a short blade and a long blade. It may be stated generally that the calculations for the turbine are made for a speed which corresponds approximately to a flow perpendicular to the inlet of a blade ring. This "calculation" speed corresponds to the optimum speed previously mentioned. For the tested converters, the speed corresponding to the maximum efficiency generally is higher than the optimum speed. This is due to the type of blading which is employed.

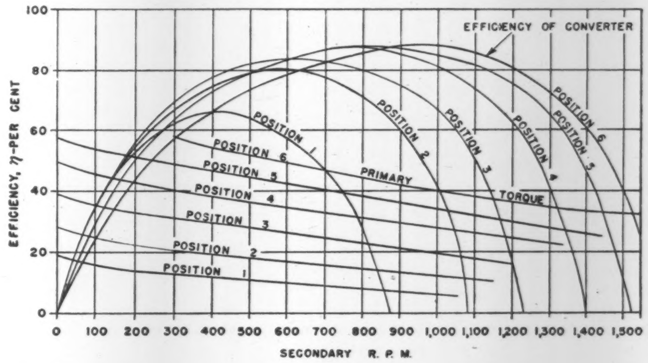


Fig. 14—Efficiency and primary torque curves for hydraulic torque converter with adjustable blades

Maximum Efficiency Speed: Speed at the maximum efficiency point will depend upon (*a*) the number of stages, (*b*) the diameter of the blade rings, (*c*) the head to be utilized, and (*d*) other factors. Obviously, the optimum speed is increased by (*a*) a smaller number of stages, (*b*) smaller diameter of stages, and (*c*) a larger head. This may be expressed also as a ratio which corresponds to the "Parsons figure" or "quality figure," as used for steam-turbine calculations, namely, $\Sigma u^2/h$, where Σu^2 is the sum of the squares of the turbine-blade velocities, and h is the head utilized in the turbine. For modern steam turbines, this ratio generally is about 2000 (metric units) for impulse turbines having some reaction, and up to 3500 for pure reaction turbines of the

Ljungstrom double-rotation type, for instance. To get a direct comparison with these figures it is necessary to multiply the corresponding values for torque converters by the metric mechanical equivalent of heat, 427. The converters now made have a Parsons figure of 3200 to 3500 at "calculation" speed, and up to 50,000 at racing.

It is possible also to use the "specific speed" of the pump and turbine as a basis of comparison. For torque converters, the specific speed n_s preferably will be expressed by the following formula

$$n_s = 3.65 \frac{n\sqrt{Q}}{h^{3/4}}$$

where n is the revolutions per minute of pump or turbine, Q is the quantity in cubic meters per second, and h is the total head in meters produced by the pump. The average value of n_s for the pump in the present converters varies between 190 to 210, and for the turbine between 150 to 160 at "calculation" speed.

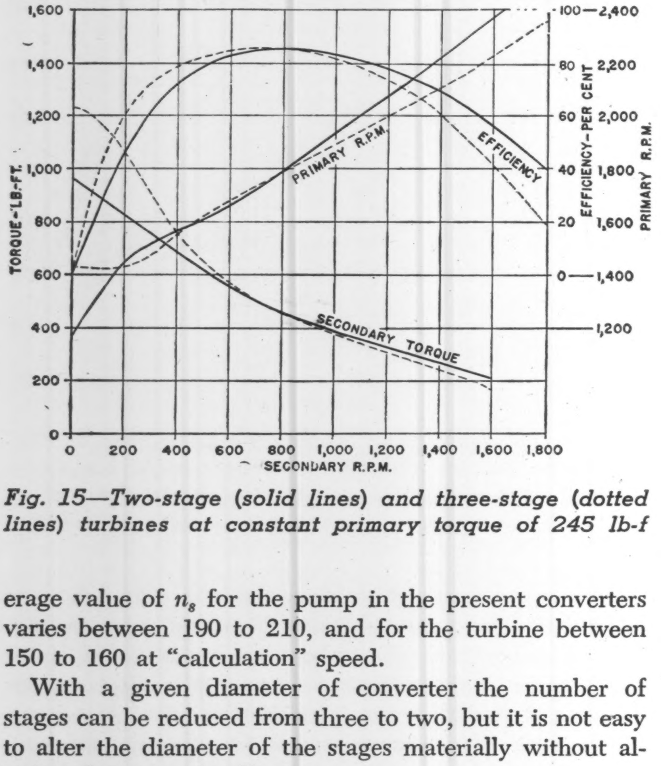


Fig. 15—Two-stage (solid lines) and three-stage (dotted lines) turbines at constant primary torque of 245 lb-f

With a given diameter of converter the number of stages can be reduced from three to two, but it is not easy to alter the diameter of the stages materially without altering the converter diameter.

Efficiency Differs for Two-Stage Unit

Some of the results of changing from a three-stage to a two-stage converter in the same casing are shown in *Fig.* 15, where it will be seen that the efficiency curve of the two-stage converter at constant torque is lower for the first 800 revolutions per minute of the secondary and, above this speed, is higher. There is also a considerable alteration in the stall ratio, that of the two-stage converter being constant at 4 to 1, and the three-stage varying from 4.5 to 1 to 5 to 1. These figures refer only to the two particular converters compared in the test. Higher stall ratios are available with different blade combinations. The effect of this on the secondary torque curve and engine speed at the commencement of converter operation is very marked.

Power-Trains Developed

PRESSING need has developed for mobile electric power-generating plants to operate base hospitals, machine shops, sanitation systems, etc., in war zone areas where power plants have been destroyed or put out of commission. The capacity required, some authorities say, is two to three per cent of the generating capacity existing in the area previous to the devastation.

Twenty-four 1000-kilowatt and ten 5000-kilowatt portable, self-contained steam-electric power plants for this purpose now are being built by Westinghouse Electric and Manufacturing Company. They are assembled in specially designed railway cars which, with suitable trucks, can be transported over most American and some foreign railroads, and can be operated to generate electric power when located on satisfactory railway sidings and provided with necessary supplementary foundation supports. This type of portable generating plant has been dubbed the "Power-Train".

Under present war emergency conditions, the primary objectives in designing and building generating plants of this type are to obtain the maximum output per unit of materials, comply with the purchaser's specifications, and meet actual operating conditions. Performance efficiency and maximum reliability of operation obtained by duplication of auxiliary apparatus and equipment must, in this case, take a position of secondary importance. The specification that the generating plant be capable of being transported over railroads in a practically complete assembled condition necessitates that the cars be arranged normally in tandem. Design is restricted further by the following operation requirements and assembly and transportation limitations:

1. Height and width clearance limitations of the railroads over which the power-trains are to be transported
2. Weight limitation per axle of the trucks and total car weight limitation on the rails
3. Full output rating to be produced when using low-grade lignite fuel with approximately 7300 Btu, 22 per cent ash, and 24 per cent moisture
4. Plants to be capable of being placed in operation without the use of an external source of power supply
5. Full-rated output of the generating plant to be obtained over an ambient temperature range of minus 40 degrees Fahr. to plus 95 degrees Fahr.
6. Cooling system to operate on limited amount of water.

Mechanical problems encountered in the design of a power-train for railway transit are in many respects fundamentally different from those associated with the power-train units in normal operation. In railway transit, the equipment must be designed with sufficient strength and be supported adequately to withstand (a) vibration resulting from vibratory forces introduced by the cars moving at freight-train speeds over rough road beds; (b) high impact forces due to sudden starts and stops; (c) large twisting forces caused by such contingencies as car derailment, etc.

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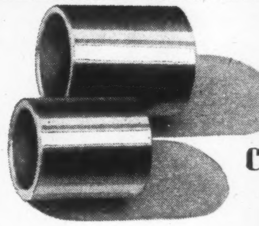
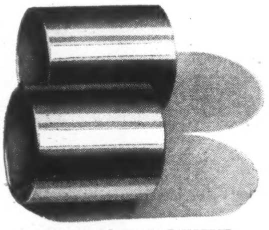
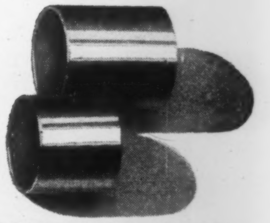


CAST BRONZE
Plain or graphited

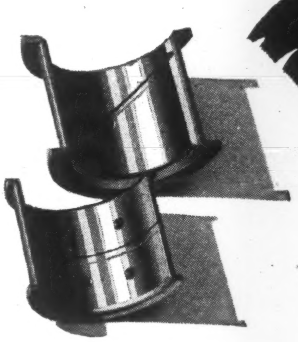


SHEET BRONZE
Plain or graphited

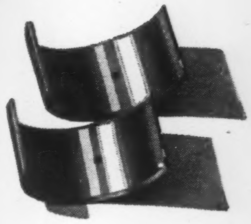
WHICH ONE SERVES BEST ?



BRONZE ON STEEL
Plain or graphited



BRONZE and BABBITT



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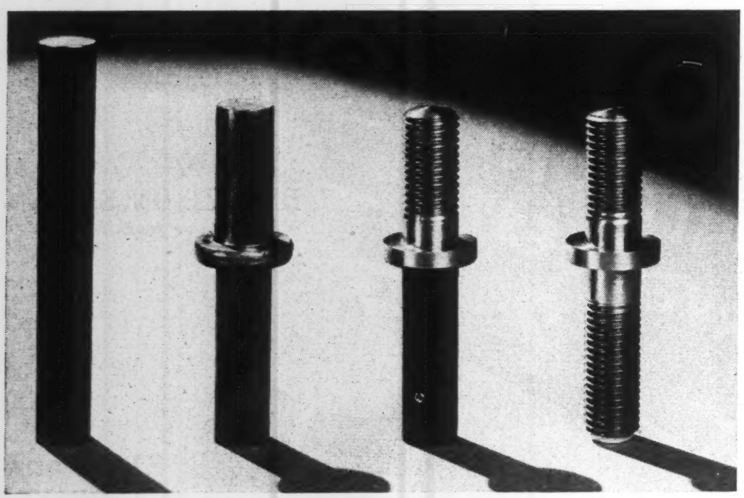
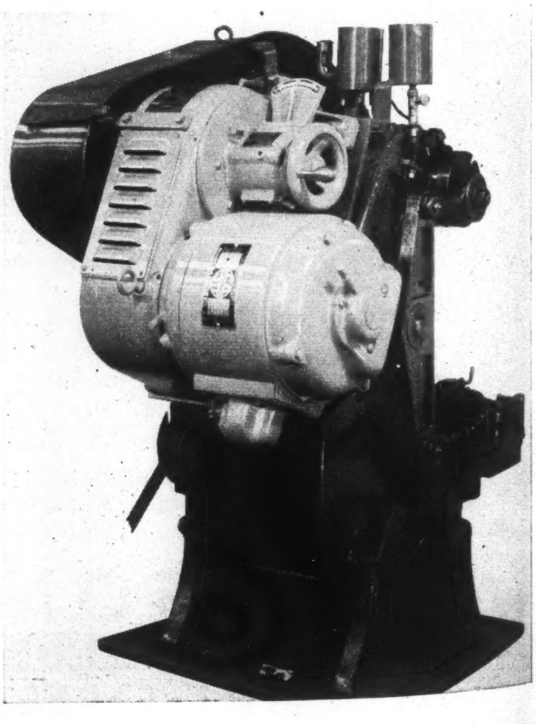


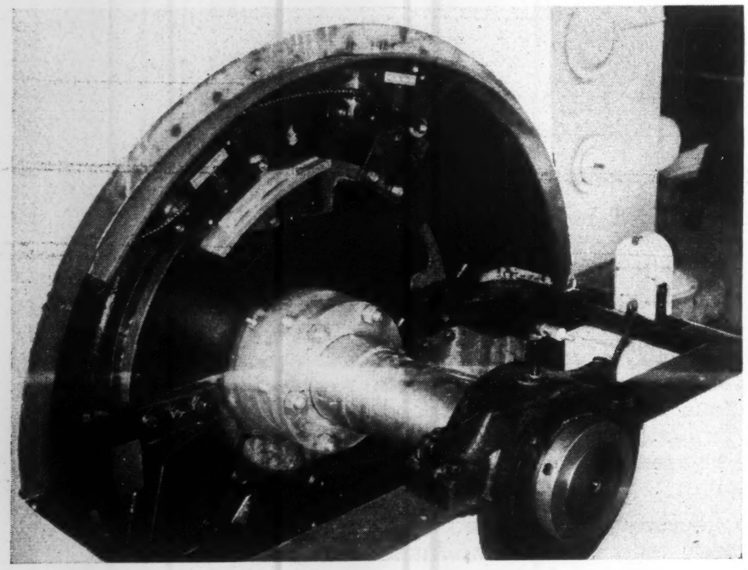
Cold Heading Saves Time, Metal

COLD-UPSET Monel collar studs, four stages in the production of which are illustrated at left, avoid the 60 per cent scrap loss involved when studs are machined from full diameter bar stock. Made by The H. M. Harper Co., the cold-formed studs show tensile strength up to 120,000 pounds per square inch with 35 per cent reduction of area. They are produced from hot-rolled rod at the rate of 50 pieces per minute with a cold header powered by a 50-horsepower motor.

Providing Correct Speeds for Press

HIGH-SPEED notching press shown below is driven by a 1½-horsepower 1800 to 900 revolutions per minute U. S. Varidrive motor. The different notching fixtures to accommodate several diameters of disks requires a variety of operating speeds, which may be set accurately when changing from one job to another. Synchronization of several presses is accomplished with only one operator.





Limit Switches Control Bakery Machine

DUMP motor which lowers and raises the mixing bowl of the Readco machine shown above is controlled by limit switches. The bowl is lowered when it is time to discharge the dough after the mixing is completed, and the limit switch on the left side of the picture stops the motor so the bowl will not travel beyond the final discharge point. When the bowl is brought back into position, the limit switch on the right side stops the motor so the bowl will stop at its proper mixing position.

wherever a tube is used...

for example **THE ELECTRONIC CALIBRATOR**

A recently designed automatic calibrator for frequency meters used in conjunction with adding machines largely eliminates tedious hand calibration, saves man hours, reduces element of human error, speeds production.

Photo Courtesy Philco Corporation

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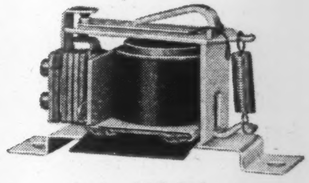
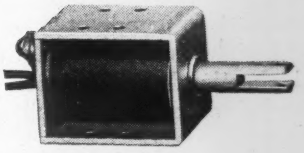


Series 120 Relay



Series 4 Solenoid

The Philco 126-tube Electronic Calibrator employs a system of fast and slow-acting relays and solenoids to bring about desired end actions. One application is the transferring of readings from the storage bank (shown above) to the keyboard of the adding machine. Operated by the plate current of OA4G tubes the relays on the storage bank energize the adding machine solenoids which press the proper number key of the adding machine.

The Guardian Series 120 relay used in this application is a small, sensitive unit having a minimum power requirement of 0.5 VA and an average of 2 VA. Coils are available in resistances from .01 to 6,000 ohms. Contact combinations up to single pole, double throw with 12.5 amp. points. Send for Bulletin 120.

The solenoid is Guardian Series 4 available for either A.C. or D.C. use. Series 4 A.C. at a maximum stroke of 1" permits a pull of 14 oz. intermittent duty, 3 oz. continuous duty. Series 4 D.C. at a maximum stroke of 1" permits a pull of 6 oz. intermittent duty, 1 oz. continuous duty. Send for information.

Consult Guardian whenever a tube is used—however—Relays by Guardian are NOT limited to tube applications but are used wherever automatic control is desired for making, breaking, or changing the characteristics of electrical circuits.

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Future Steels

(*Continued from Page* 132)

3. Grain size and normality (influenced by steelmaking practice and by contents of obscure elements)
4. Presence of "special addition agents" if any.

To establish fair specification ranges, we must know how much spread in hardenability users can tolerate. As an initial step, a joint A.I.S.I.-S.A.E. committee now is trying to ascertain how much spread has been involved in the past, in steels of present standard composition ranges. Of interest in these studies are such cases as shown in *Fig.* 4, which shows total spread in hardenability found by Chevrolet in 30 heats of N.E. 8620 steel, compared with the spread Chevrolet considers desirable (in this or any other steel) for use in certain truck transmission gears.

For many parts it is necessary only to specify minimum required hardenability. This problem is relatively simple, and considerable steel already is being purchased on that basis. Other applications will require specifying definite ranges of hardenability—minimums to assure hardening of sections involved and maximums to reduce quench-cracking, to prevent hardening too deeply or to too high hardness, or to control weldability. As an example, seven heats of steel of nearly identical chemical compositions, one group of three treated with one special addition agent, and the other group of four with a different, more potent agent, were rolled into ½-inch thick plate, fabricated into a certain part, hardened and tempered to brinell hardness number 341-388, and assembled by automatic arc welding. Hardenabilities of the two groups of steels are shown in *Fig.* 5. All these heats had sufficient hardenability to respond as desired to heat treatment. The higher hardenability group, without exception, cracked badly along the overhanging plate edge after being welded into the joint shown in *Fig.* 6 (*a*). No cracking or other difficulty was experienced with the lower hardenability group of heats during welding into the same design joint by the same process. It should be noted that near the left ends of the hardenability curves, indicating suitability for hardening in ½-inch plate sections, the two groups are practically identical. The significant difference from the standpoint of cracking near welds is farther to the right, in the slower cooling zone. It was possible in this instance to modify the admittedly critical joint, as shown in *Fig.* 6(*b*) so as to permit using the higher hardenability steel; but this incident is illustrative of need for controlling hardenability within a range.

Specifying Steel by Hardenability

It now appears unquestionable that hardenability specifications are coming in the not distant future. The few large users now purchasing some of their steels to hardenability specifications will continue and enlarge this practice as experience accumulates on certain parts. At present, steel producers are apparently not in position to control hardenability consistently within the narrow ranges some users will specify; hence outlets for "off heats" must be available, either to the same or another user who can utilize hardenabilities slightly above or below the range aimed at, or to small users or warehouses buying only to chemical composition specifications. Therefore, general use of hardenability specifications may at first be confined to widely used standard steel types.

Work on "special addition agent", "needled", "additive", "treated", or "intensified" steels (we still haven't a good name for them) started nearly ten years ago. Use of and knowledge concerning the addition agents developed slowly, with a few large users (notably Buick and Timken-Detroit Axle) pioneering in experimental and production applications.

Needled steels are made, usually in open-hearth furnaces, by normal practices, each heat first being thoroughly deoxidized by conventional means and a minute percentage of one of the special agents then added, in ladle or ingot mold. The agents are made and sold by various alloy manufacturers and have one feature in common—all contain boron. The greater the percentage of boron present in an agent, the smaller the proportion of agent added to steel; it should not be concluded from this, however, that boron is the only active ingredient.

The most important and the only generally accepted effect of adding any one of these agents to properly prepared steel is increase in hardenability; an example is shown in *Fig.* 7 (19). All compositions do not respond alike, higher manganese steels, for example, showing relatively greater hardenability increases than do lower manganese steels of otherwise similar compositions.

Special Addition Agents Effect Great Savings

Additive steels have been and are used in large quantities in many production applications, including aircraft and ordnance armor, axle shafts, rear-axle gears, steering knuckles, and steering arms. Some of the savings made in alloying elements are actually startling, although the following examples do not necessarily imply that original and replacement steels have equal hardenabilities (symbol "T" in middle of steel number is used to indicate "treated" steel):

1. 10T40 for 3240 and 4340 in axle shafts
2. 40T23 for 4620 in rear axle drive gears
3. 10T24 for 4815 in rear axle drive pinions.

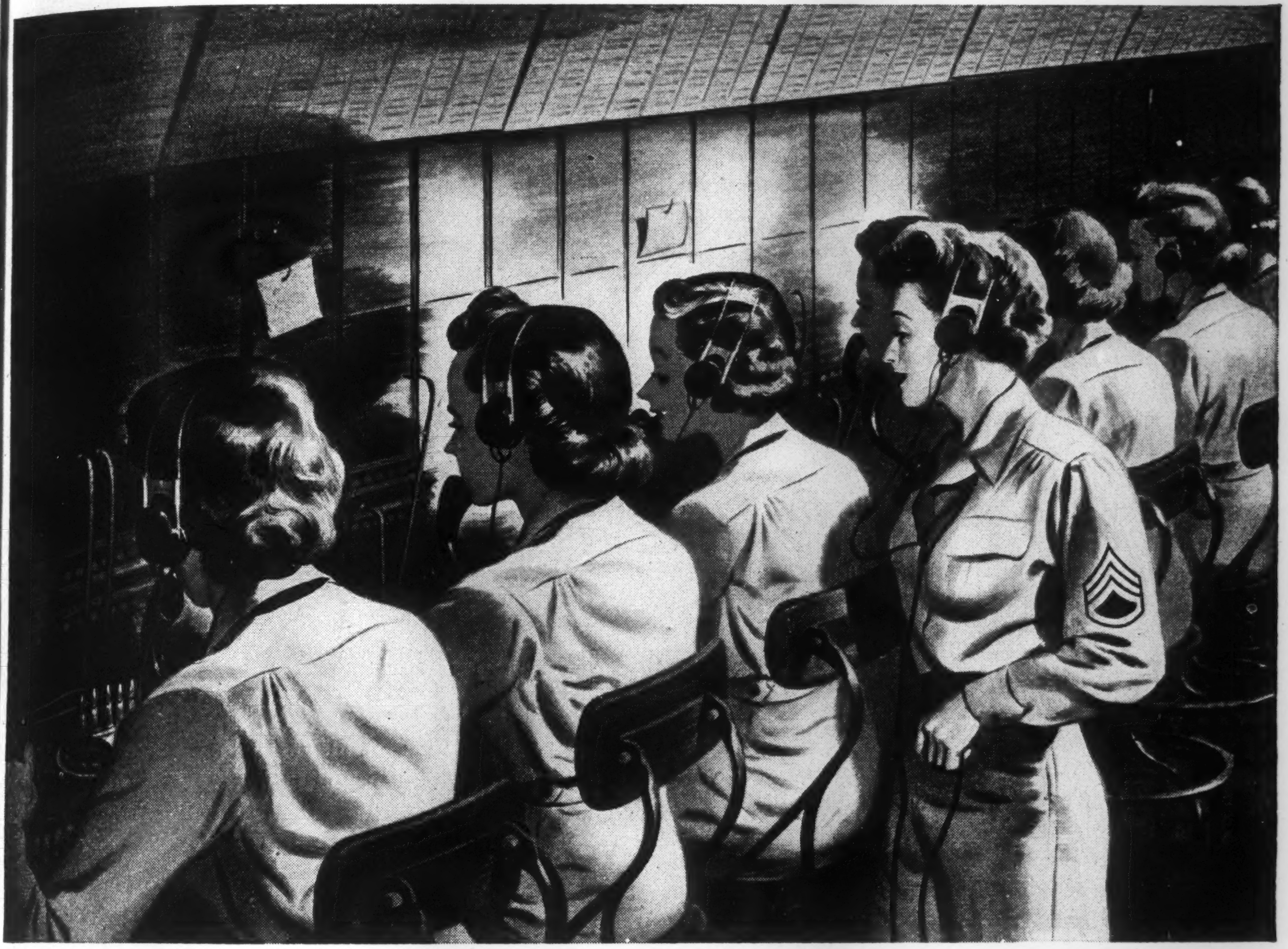
From specification standpoint, U. S. Army Ordnance has led the way with tentative specification AXS-930 (20), based on hardenability and strength-toughness relationships, and covering five types of needled steels. It is probable that the future will see additional user specifications for such steels, based primarily on physical test requirements.

Formulation and adoption of N.E. alloy steels resulted, of course, solely and directly from the war, because of scarcity of alloying elements. Compositions were worked out with three concepts in mind:

1. Hardenability of each composition should be closely comparable to that of a long-established, well known and widely used alloy steel, if possible of equivalent carbon content.

2. Minimum additions of virgin alloying elements must be required, and no residual alloying element wasted.

3. Mechanical properties of all alloy steels heat treated to identical structures and hardnesses should be alike. That this is true for all hardness ranges is not generally



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admitted. It appears a safe assumption up to about brinell hardness of 400.

The N.E. types most widely used and now being promoted in preference to all others for automotive and other structural applications are the nickel-chromium-molybdenum or so-called three-alloy types. These steels may be briefly compared with older peacetime alloy steels in some of the more significant properties as follows:

1. Relationship of Toughness to Hardness. Complete data have not been obtained, but the great majority of available data indicate no appreciable difference. A few exceptions have been reported—for example, greater brittleness of NE 9420 than SAE 4320 in case-hardened transmission gear teeth—so that some metallurgists have reserved judgment concerning toughness of N.E. steels at high hardnesses.

Promise Improved Machinability

2. Machinability. Here again, in most cases no appreciable difference can be found. A few instances of slightly inferior machinability of N.E. steels, such as spiral bevel gears after annealing and axle shaft splines after heat treating to high hardness, have been reported. Further work on annealing cycles perhaps will enable users to produce machinabilities in N.E. steels equal in all applications to those of comparable S.A.E. alloy steels.

3. Carburizing Characteristics. Ratios of depths of cases to carburizing times for N.E. steels appear to be of the same order as for other alloy steels. Preliminary investigations have indicated, however, that at lower temperatures (1650 degrees Fahr.), pack-carburizing produces appreciably higher carbon concentrations in outer portions of cases of three-alloy N.E. steels than in nickel-molybdenum steels, both after direct quenching and after box cooling. At 1700 degrees Fahr., differences after direct quenching are so small as to be negligible, although N.E. steels are apparently less subject to the decarburizing action, during box cooling, characteristic of some other steels. This property, if shown to be consistent by results of more extensive tests now under way, may be an important reason for using three-alloy steels for slow cooling after carburizing in solid compound.

4. Hardenability Spread. As use of hardenability as a primary requirement grows, there should develop a trend toward selection of composition types which inherently fall within narrower hardenability ranges. Under present conditions, lack of alloy-free scrap means waste if single or two-alloy steels are used. After the war the situation in some steel mills may be such that single-alloy steels can be made without appreciable residual unspecified alloying elements; but other mills may, for several years at least, find it impossible to make large quantities of low-residual steels. Since we still are without sufficient test data to establish hardenability spreads authoritatively, the subject must for the present be studied by calculation. We shall use the Field method (18), confining our examples to medium-hardenability steels for which the method is known to be reasonably accurate, and, in the interest of simplification, eliminating for the present purpose effects of grain size (using No. 7 in all cases), of phosphorus and sulphur (which nearly cancel each other), and of residual vanadium and copper, since all these effects are the same for all steels here considered. Hardenability of each steel is calculated three ways:

A—Each element at minimum of specification range, with no unspecified nickel, chromium or molybdenum.

B—Each element at maximum of specification range, with no unspecified nickel, chromium or molybdenum.

C—Each element at maximum of specification range, with each unspecified element at maximum permitted by A.I.S.I. standards (nickel .25 per cent, chromium .20 per cent, molybdenum .06 per cent). In steels containing specified amounts of nickel, chromium and molybdenum, calculations B and C are identical.

Table II lists compositions and calculated Grossmann critical diameters of the .40 per cent carbon steels here considered. Calculated A, B, and C hardenability curves for the S.A.E. one-alloy, S.A.E. two-alloy, and N.E. three-alloy steels are shown in *Figs.* 8, 9, and 10, respectively. They indicate clearly that, in absence of residual unspecified elements, hardenability spreads are roughly proportional to the number of alloying elements contained, also that ranges widen in proportion to number and percentages of unspecified elements present. Of unspecified elements, the present maximum permissible content of chromium is by far the most potent, with molybdenum next.

Contributions from the following are gratefully acknowledged: J. O. Almen, C. O. Ball, E. V. Bennett, J. M. Diebold, G. C. Farnsworth, W. F. Herdrich, E. O. Mann, R. H. McCarroll, R. C. Nelson, F. L. Racine, C. W. Roland, R. W. Roush, R. B. Schenck, L. L. Schiering, H. G. Taracks and F. C. Young.

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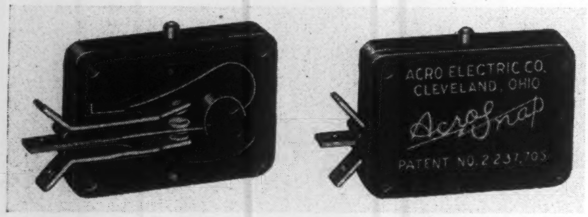
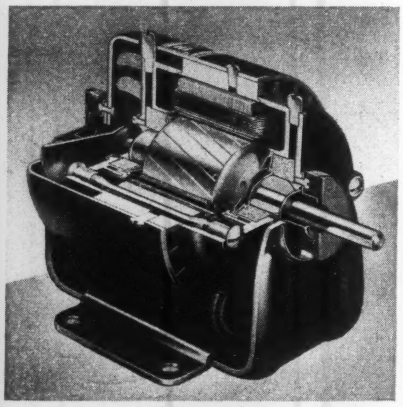
Miniature Snap-Action Switch

CLAIMED TO BE the smallest fully-enclosed snap-action switch built, the new Miniac is 17/64-inch thick, 13/16-inch high and 1 3/16-inch long. Developed by Acro Electric Co., 1311 Superior avenue, Cleveland 14, the switch is fully enclosed in a plastic case with four mounting holes 3/32-inch in diameter. Actuation is provided by a stainless steel pin plunger. The main blade, contact blade, and roller spring are beryllium copper. Contacts are of fine silver and all other parts are noncorrosive. The switch is rated at 15 amperes, 115 volts alternating current, and is furnished in single-pole, normally open and normally closed, double-throw types. It is designed to permit leaf type or overtravel plunger type actuators to be attached to the case.



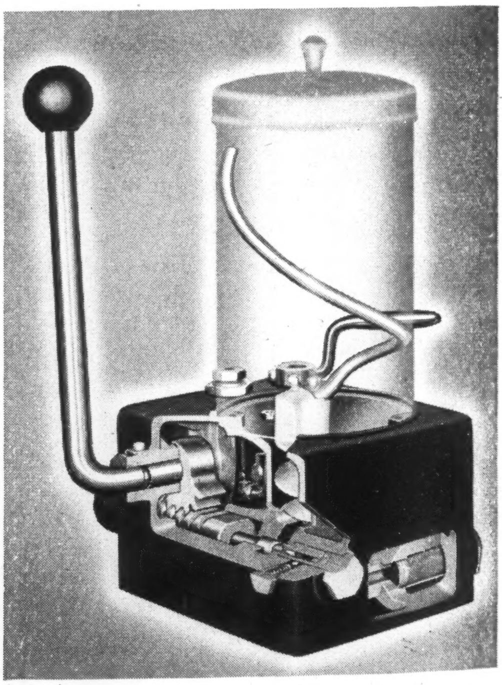
Motor with Pressure-Locked Mounting

INCORPORATING a "pressure-locked" rubber mounting, the new Type "T" shaded-pole Micromotors offered by A. G. Redmond Co., Owosso, Mich., are for power requirements of 1/125 to 1/20-horsepower. This new pressure-locked method provided on base-mounted models requires two large resilient rubber pads, two spring steel lock rings, and a base welded into a single rigid unit. Quiet operation is achieved due to the generous amount of free-acting resiliency. The company's "flush-weld" rotors are used in the Type "T" Micromotors. All rotor slots are copper-filled flush with circumference, with preformed copper inductors welded permanently into position for maximum electrical conductivity in the circuit. An air-stream innovation of enclosed cooling fins and fan permits the use of a much smaller sized motor than is possible in ordinary shaded-pole motors of comparable power. Frames are of three-piece die-cast design. Nongumming porous bronze bearings, spring-tension centered, filter the lubricant, and are protected by large capacity oil reservoirs. Stator windings, which are impregnated with a special varnish, are unaffected by oil or humidity.



Centralized Lubrication Systems

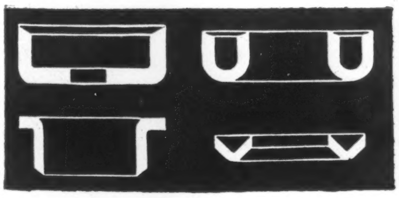
AMONG THE FOUR new centralized lubrication systems for light and heavy industrial use, announced by the Alemite Division, Stewart-Warner Corp., 1826 Diversey Parkway, Chicago 14, is the Dual Manifold system. This system is designed for applying high-pressure grease or oil lubrication to an unlimited number of bearings in equipment exposed to especially heavy-duty or outdoor conditions. It is a simple, direct hydraulic system by which a fully adjustable predetermined charge of any



This NEW HANDBOOK—

Chapter VI
COMMON TROUBLES ENCOUNTERED
EFFECTS OF EXCESSIVE CLEARANCE
A HANDBOOK ON
HYDRAULIC & PNEUMATIC
LEATHER PACKINGS
E. F. HOUGHTON & CO.

...will help you on Packing problems



TYPES AND USES

VIM Leather Packings are supplied in "Cup," "Flange," "U" and "V" form, also special shaped washers and gaskets. Outstanding war applications include sealing of hydraulic mechanisms in many places in modern airplanes, recoil mechanisms of guns, presses, jacks, steel mills and machine tools—for all types of hydraulic and pneumatic seals.

No subject is of more pertinent interest to plant or design engineers than proper operation of hydraulic and pneumatic equipment. This 176-page handbook was prepared to help men understand the principles of good packing design, and how to avoid trouble.

You can obtain a free copy of the revised 1944 edition by requesting it on your firm's letterhead. Write—

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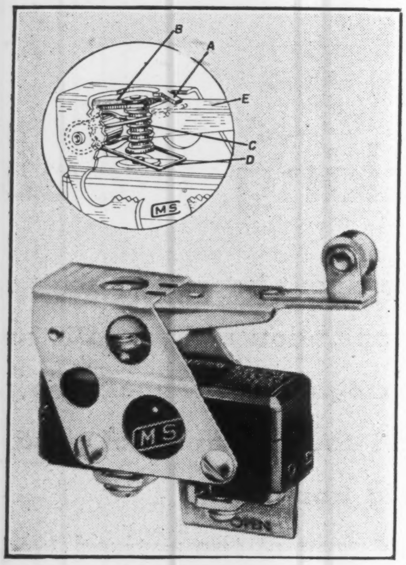
type of grease or oil at any temperature may be introduced into all bearings of a machine. Dual manifold valves are available with two, six, eight or ten outlets. Model 7040 stationary hand-operated pump, with 5-pound lubricant capacity, is featured with the system for permanent mounting into machines. The accompanying illustration shows this pump. The other three systems announced by the company include the LubroMeter system for delivering oil or grease to an unspecified number of machine bearings simultaneously through a single line which is serviced by hand or power-operated portable type guns or by fully automatic equipment. The Dual Progressive system similarly supplies an unspecified number of bearings through a single line, but one at a time; and the third—the Progressive system—supplies grease to each of 3 to 20 bearings through 3 to 20 individual lines. The systems are applicable to small, medium or large machinery.

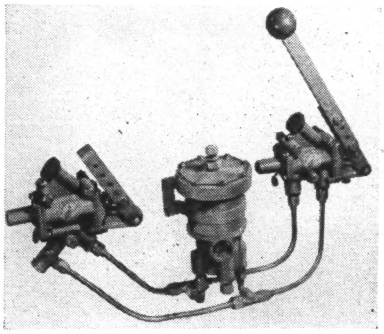
Adjustable Actuator Bracket

DESIGNATED AS AD-5721-R a new adjustable actuator bracket introduced by Micro Switch Corp., Freeport, Ill., permits adjustment on the job of both the operation point and pretravel. It retains the sealed-in features of the basic Micro switch which prevent tampering with the elements that control the characteristics of the switch, but provides tabs on the steel actuator bracket and the lever arm which may be bent to change the mechanical characteristics of the unit. By bending Tab "A" downward the lever mechanism is held nearer to the operating plunger of the R31 Micro switch. The more the tab is bent, the less the pretravel. After each operation the lever arm is returned to free position by the pretravel spring "D" which rests on top of the switch unit. By bending Tab "B" up or down, the operating point of the combination is lowered or raised. This tab is a part of the steel lever arm "B" and carries the overtravel spring mechanism "C" consisting of a spring which is preloaded above the operating point of the switch unit so that it compresses only after the switch operates. Lever "E" carries the motion of the actuator's mechanism. The R31 switch for which this actuator was designed is easily and quickly assembled into the steel bracket, which also affords mechanical protection to the switch.



Remote Control System

DEVELOPED originally for use in large aircraft, ISO-draulic remote controls announced by Adel Precision Products Corp., Burbank, Calif., are now being offered design engineers in railroad, marine, machine tool and other industrial fields. Combining simplicity of design with flexibility of use, the control system operates efficiently in temperatures ranging from 160 to −65 degrees Fahr., and remains in synchronism through temperature changes and/or with sections of the system at different extremes of the temperature range. The diaphragm and auxiliary chamber of the equalizer unit have been eliminated. Synchronizing screws by-passing the pistons of the master and slave units permit bleeding and filling the system at one point without breaking any line connections, and provide rapid indexing of lever positions. The two-line system provides smooth accurate response at the slave unit to motion in either direction initiated at the master unit. It is available with or without an integral lock which in the slave unit of the irreversible system maintains the selected position against load and vibration. Locking system and piston design have been improved to reduce the handle load to 4½ pounds for the irreversible system and 2½ pounds for the reversible system, using an 8-inch lever. The units can be mounted in any position and at varying elevations. Stacking width of master and slave units is 1¼ inches. Total weight is less than 3¼ pounds.



New Inlay Process Developed

FOR PLACING characters on metal panels, chassis, etc., a new inlay process announced by Screenmakers, 64 Fulton street, New York 7, eliminates the use of nameplates on front panels. Application of the process is carried out in either a flat or a wrinkled background, on finished metal or on metal in the bare state. Backgrounds can be black, olive drab, brown, or any shade desired. Characters, inlay-baked, are white, red, yellow, green, or any required

VERSATILE HANSEN COUPLINGS...

HANSEN COUPLINGS are versatile, in that they are made for a wide range of uses, such as for air, oil, grease, oxygen, gasoline and acetylene. Designed, engineered and built to handle air, liquids and gases in the most efficient manner with absolutely no losses whatever due to leakage. Hansen couplings will safely handle pressures from 2 ounces to over 10,000 pounds.

Extremely easy to operate, merely pusn plug into socket, it is locked absolutely tight, slight pull on sleeve, it is unlocked and detached. When plug is inserted in the air, oxygen, or acetylene couplings, it automatically opens the valve and automatically closes it when plug is ejected . . . No twisting, turning or locking, complete swivel action eliminates kinking of hose line.

Large illustration at left shows Push-Tite coupling which handles air, oil, and grease while couplings illustrated below handle gas, acetylene and oxygen.

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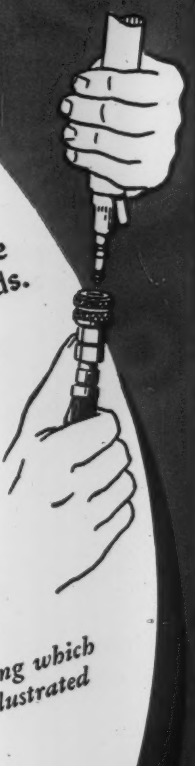

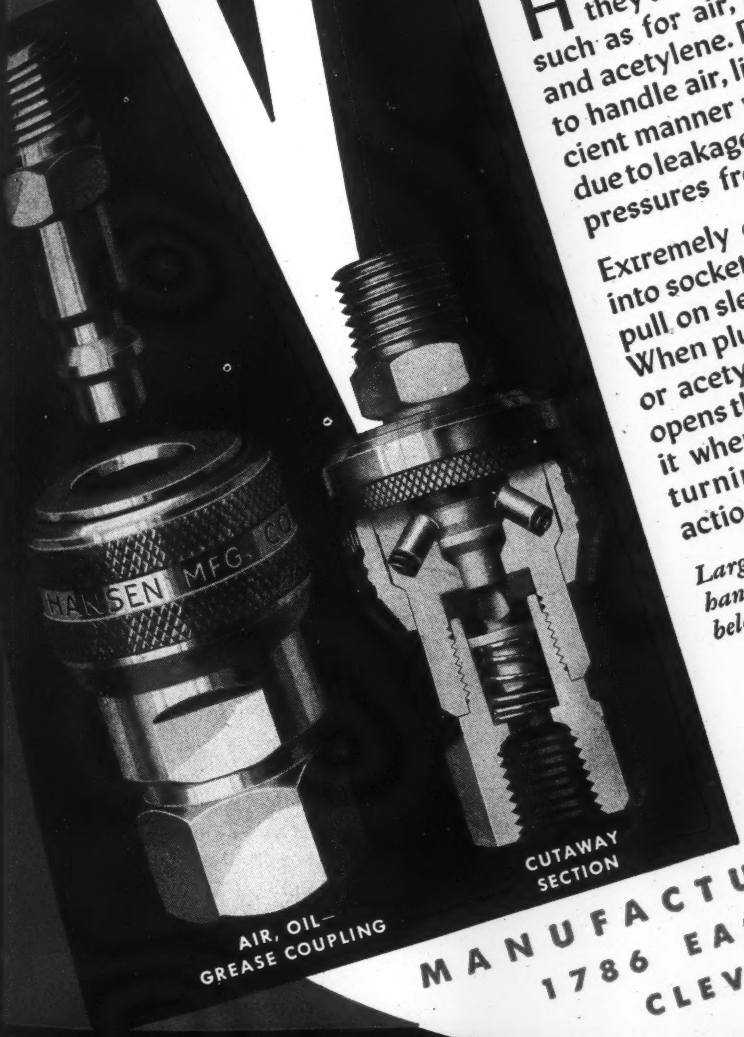

AIR, OIL—
GREASE COUPLING

CUTAWAY
SECTION

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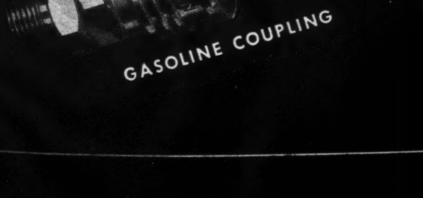
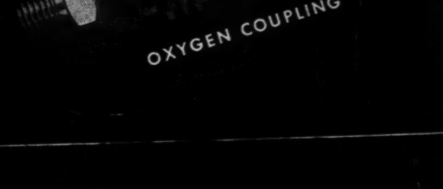


GASOLINE COUPLING

ACETYLENE COUPLING



OXYGEN COUPLING

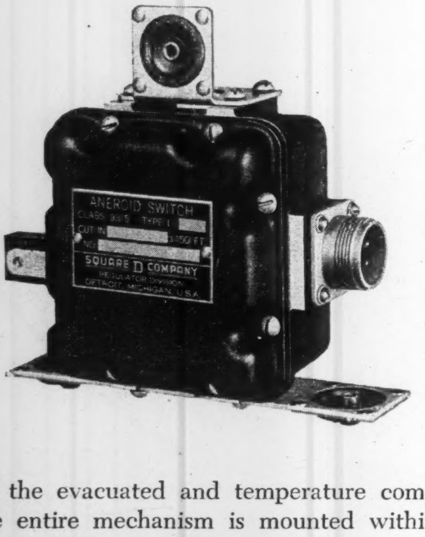
color. The background finish protects the inlaid characters and makes them resistant to abrasions and salt spray. The processed characters are guaranteed to pass the 50-hour salt spray test. The inlay is produced with enamel, silk-screened onto the metal. It is then masked and a pebble or a smooth metal background in the color desired, is sprayed on the metal. The thickness of the background gives the original lettering its inlay effect and protects same. It is then baked. Attaching of the inlay is done by the company, while the fabricated metal (raw or finished) is furnished by the manufacturer with directions or blueprint, or both, as to the characters to be inlaid.

Hydraulic Aircraft Relief Valve

FEATURING SIMPLICITY of design with a minimum number of parts the new valve for airplane hydraulic systems, announced by Pesco Products Co., 11610 Euclid avenue, Cleveland 6, meets the rigid AAF winterization requirements. Tests have proved high operating efficiency through a temperature range of −65 to 160 degrees Fahr. Models in various sizes are now available in accordance with AN specifications.

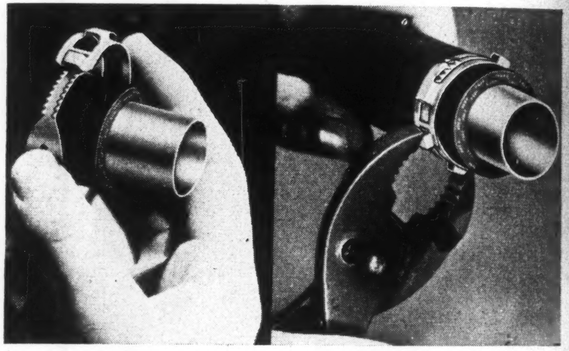
Bellows-Operated Pressure Switch

SUITABLE FOR use over a temperature range of −65 to 160 degrees Fahr., Class 9315 Aneroid switch has been added to the line of aircraft electrical control devices produced by the Square D Co., Regulator division, 6060 Rivard street, Detroit. The switch has been designed for close control of engine superchargers but offers possibilities in other applications involving electrical control responsive to altitude change. It comprises a precision switch mechanism actuated by a sensitive diaphragm or bellows of the evacuated and temperature compensated type. The entire mechanism is mounted within a die-cast, anodized, aluminum housing provided with gasketed cover sealing the internal mechanism from the atmosphere. Manifold or atmospheric pressure connection is through a ⅛-inch tapped hole. The unit is provided with an approved two-prong, pressureproof fitting for plug-in electrical connection to the switch. A three-point, shock-proof mounting designed to minimize vibration and shock effects is provided. Operating range for the switch is up to 35,000 feet with a minimum differential of 1000 feet. Net weight is 24 ounces. At 28 volts direct-current, the switch will handle up to 25 amperes in-rush current on solenoids, etc.



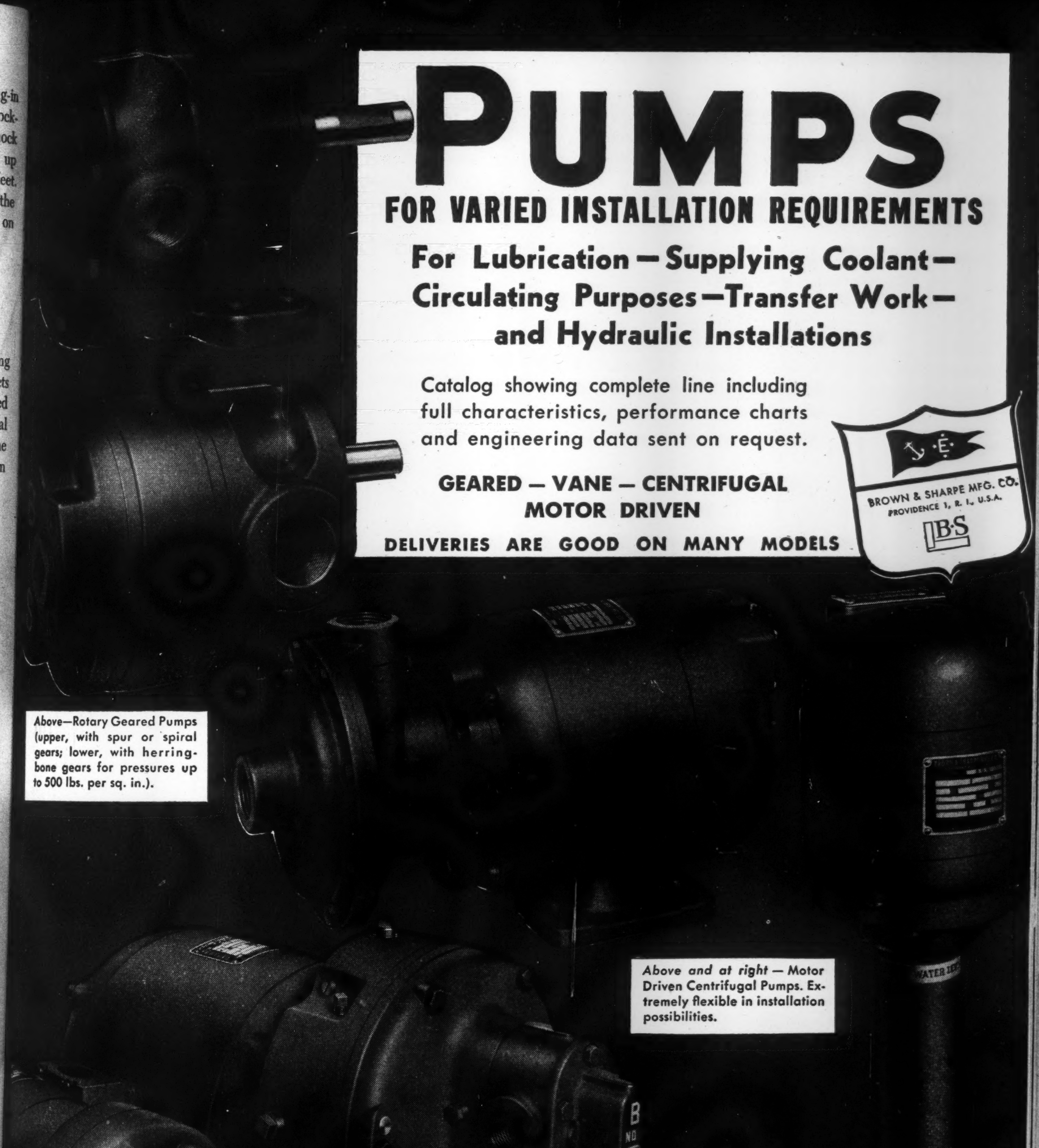
One-Piece Hose Clamp Offered

USING NO GEARS, thumb screws or intricate locking means, the new clamp introduced by Tinnerman Products Inc., 2085 Fulton road, Cleveland 13, may be snapped over a hose into a prelatched position by hand. Final lock is made with hand pressure on ordinary pliers. The clamp has a low profile, is light in weight and exerts even pressure around the circumference of the hose. It is made for low-pressure connections in one piece, using no threaded parts. This reduces weight and assembly time. Made of S.A.E. 1060 spring steel with Parkerized and zinc chromate primer finish, the clamps are available for all sizes of AN and Ordnance specification hose in a wide range from ½-inch outside diameter and up.



Fuel Oil Clarifiers for Diesels

AN IMPROVED standard line of round tank fuel oil filters for diesel engines has been announced recently by Briggs Clarifier Co., 1339 Wisconsin avenue, Washington 7. Addition of these new models has broadened the application of the company's fuel oil clarifiers so that flow capacities range up to 500 gallons per minute. Maximum working pressures and hydrostatic test pressures have been established to meet specific demands of applications. For small high-speed diesel engines where pressures may run above 50 pounds per square inch, the clarifier is designed for a maximum working pressure of 100 pounds per square inch, and is hydrostatically tested at 150 pounds per square inch. For large, heavy-duty diesels where pressure is usually between 15 and 25 pounds, it is designed for a maximum working pressure of 40 pounds per square inch, and hydrostatically tested at 60 pounds per square inch. Pressure drop across the refills ranges from 0 to 5 pounds per square inch on all models. Covers and cover gaskets are interchangeable in the various sizes, while



PUMPS

FOR VARIED INSTALLATION REQUIREMENTS

For Lubrication — Supplying Coolant — Circulating Purposes — Transfer Work — and Hydraulic Installations

Catalog showing complete line including full characteristics, performance charts and engineering data sent on request.

GEARED — VANE — CENTRIFUGAL
MOTOR DRIVEN

DELIVERIES ARE GOOD ON MANY MODELS

BROWN & SHARPE MFG. CO.
PROVIDENCE 1, R. I., U.S.A.
B·S

Above—Rotary Geared Pumps (upper, with spur or spiral gears; lower, with herring-bone gears for pressures up to 500 lbs. per sq. in.).

Above and at right — Motor Driven Centrifugal Pumps. Extremely flexible in installation possibilities.

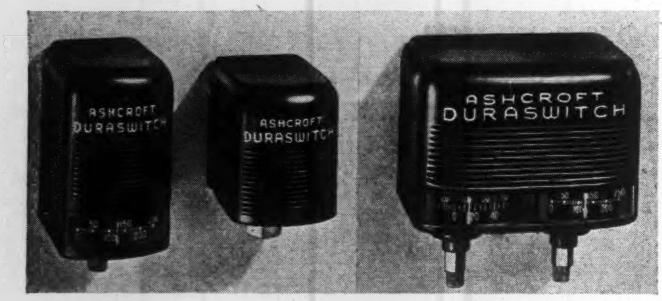
At left — Motor Driven Rotary Geared Pumps. Installation convenience is distinct advantage.

BROWN & SHARPE

spare gaskets may be ordered with bolt holes already punched. These gaskets are made of a mixture of cork and synthetic rubber. In the refill holding assembly, improvements have been made to assure a seal at both ends of the refill cartridge. In addition to the complete standard line of fuel oil clarifiers, custom-built models also are available to meet specific requirements.

Pressure and Temperature Switch

OFFERED IN two series Nos. E-100 and E-200 by Manning, Maxwell & Moore Inc., Bridgeport 2, Conn., the Duraswitch is a sturdy, dependable switch in combination with the Ashcroft gage. The gage has a slide rule dial easily readable for accurate adjusting of the switch to control point. This combination saves space on equipment, eliminates extra fittings, saves on installation and contributes toward improved appearance. The series E-100 is for pressure only, while the E-200 is for both pressure and temperature applications, as a pressure control and pressure gage combination, or temperature control and thermometer combination, in single or duplex types.



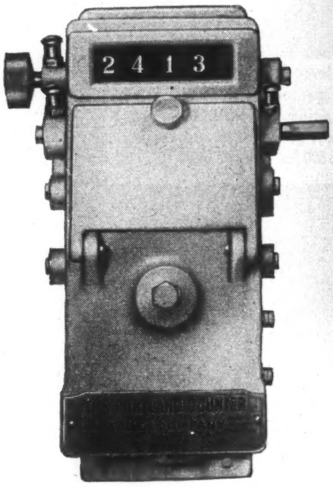
Hydraulic Drilling Units

COMPLETION OF a new special hydraulic drilling unit for drilling, reaming, facing, boring, counterboring or tapping, in any one position or a combination of horizontal, vertical or angular positions, has been announced by the Barnes Drill Co., Dept. 79, 814-30 Chestnut street, Rockford, Ill. It is available in three sizes (Nos. 5, 10 and 20), motorized according to requirements. The No. 5 unit is equipped with a 5 to 7½-horsepower motor; No. 10 with a 10 to 15-horsepower; and No. 20 with a 20 to 25-horsepower. The latter two have the essential parts built-in complete with a single motor, driving the spindle and hydraulic pump for any automatic hydraulic feed cycle. The No. 5 unit is equipped with a separate motor-driven hydraulic system. The units can be used horizontally, either singly, grouped around an indexing table, or in combination with vertical and angular applications. One standard unit, equipped with an 8-spindle auxiliary head and used horizontally as a self-oiling production machine, is capable of facing and counterboring as many as 8 bosses simultaneously. A machine equipped with two drilling units, in an angular position, may be used for drilling and counterboring as many as 12 holes, two at a time. The hydraulic system of the built-in motor on the Nos. 10 and 20 units is automatically cooled by a fan blower.

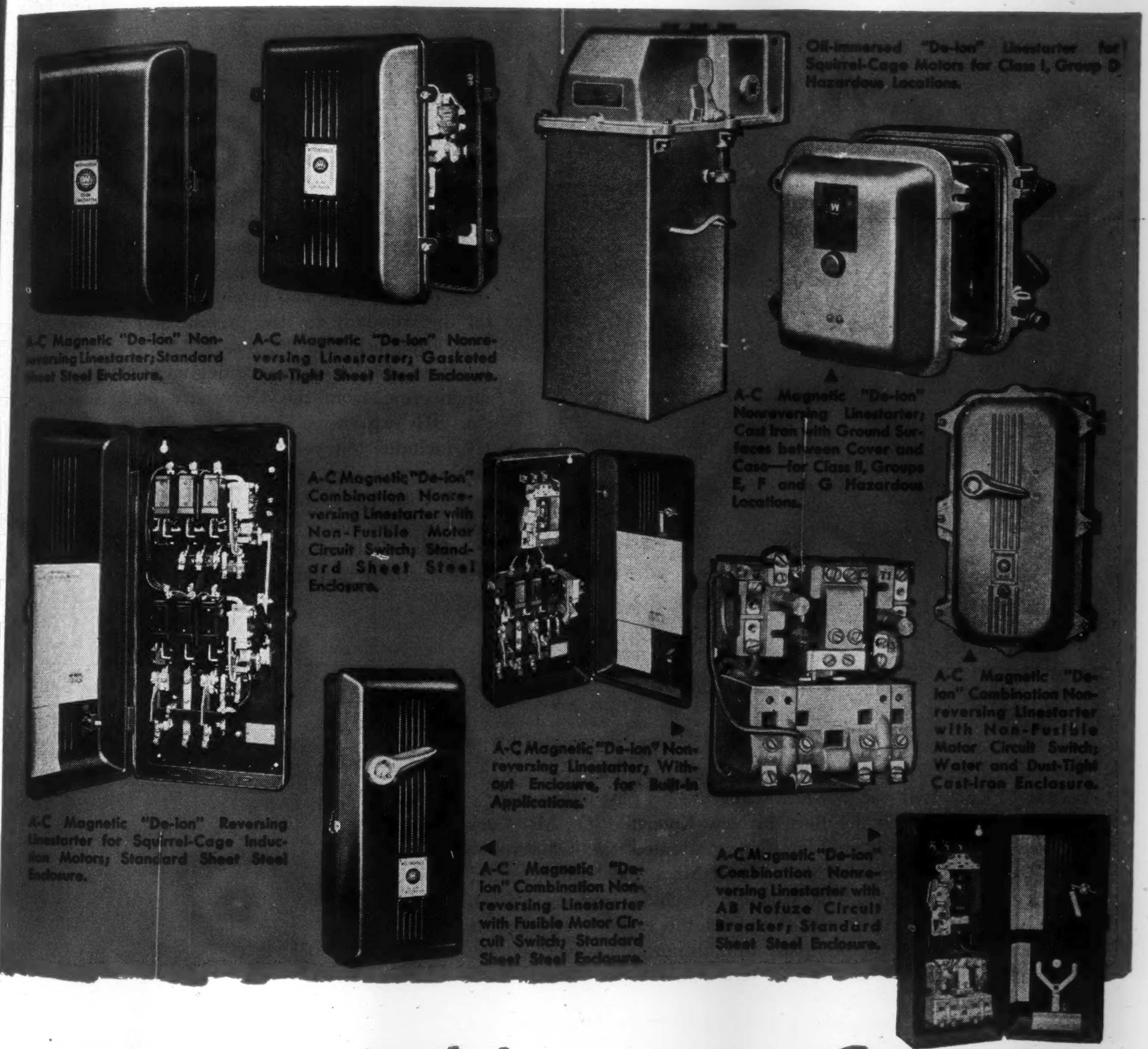
Continuous Strap Clamp

VIBRATIONPROOF regardless of conditions or length of service, the new hose clamp of Actus Products Co., Mount Vernon, N. Y., incorporates two straps, a new housing, stronger screw and increased take up. Flexibility is retained and strength has been added. Designed to meet requirements for gas, oil and coolant lines where pressure is a vital factor, the continuous strap clamp may be attached in numbers, securing any length and maximum takeup. Electrogalvanized strap in ¼-inch, ⅜-inch and ½-inch widths made to order in stainless steel will conform to any shape. The strap is placed around the hose and the end is inserted in a slot at the bottom of the buckle. It is then pulled through the slot and all slack is taken up before the thumb-screw is tightened. The double strap clamp is furnished in 5, 8 and 12-inch sizes, fitting hoses from ⅜-inch to 3½ inches in outside diameter.

Predetermining Control Counter



FEATURING AN automatic repeat mechanism that an operator can set at any desired number and which automatically repeats in units of the set count without further attention, the new control counter announced by Adde & Co., Portland, Me., is a rugged, heavy-duty unit. At the completion of each set count a 110-volt switch is actuated to control the operation. This switch may be furnished for manual reset or an automatic type reset. The manual reset requires the depressing of a pushbutton at the lower end of the case, while the automatic type reset requires the counter to be operated through one count before being automatically reset, or in a 110-volt solenoid reset for remote control. When the desired quantity has been tallied, alarm signal contacts are closed during one count and the electric control switch actuated. The switch may be furnished to make or break contact at the completion of each preset count and may be used to stop the machine or to start other operations. Enclosed in a heavy dustproof case, the control counter is adapted for speeds up to 400 counts per minute. The electric switch is rated 110 volts, 10 amperes. For higher capacities a starter switch or relay is necessary. Terminals are provided for alarm signals for either light or bell operating on 6-volt current.



A-C Magnetic "De-ion" Nonreversing Linestarter; Standard Sheet Steel Enclosure.

A-C Magnetic "De-ion" Nonreversing Linestarter; Gasketed Dust-Tight Sheet Steel Enclosure.

Oil-immersed "De-ion" Linestarter for Squirrel-Cage Motors for Class I, Group D Hazardous Locations.

▲
A-C Magnetic "De-ion" Nonreversing Linestarter; Cast Iron with Ground Surfaces between Cover and Case—for Class II, Groups E, F and G Hazardous Locations.

A-C Magnetic "De-ion" Combination Nonreversing Linestarter with Non-Fusible Motor Circuit Switch; Standard Sheet Steel Enclosure.

▶
A-C Magnetic "De-ion" Nonreversing Linestarter; Without Enclosure, for Built-In Applications.

▲
A-C Magnetic "De-ion" Combination Nonreversing Linestarter with Non-Fusible Motor Circuit Switch; Water and Dust-Tight Cast-Iron Enclosure.

A-C Magnetic "De-ion" Reversing Linestarter for Squirrel-Cage Induction Motors; Standard Sheet Steel Enclosure.

◀
A-C Magnetic "De-ion" Combination Nonreversing Linestarter with Fusible Motor Circuit Switch; Standard Sheet Steel Enclosure.

▶
A-C Magnetic "De-ion" Combination Nonreversing Linestarter with AB Nofuze Circuit Breaker; Standard Sheet Steel Enclosure.

WHEN YOU NEED *Magnetic Controls*

FOR SINGLE-PHASE AND SQUIRREL-CAGE MOTORS CONSULT WESTINGHOUSE BUYING DATA (NEW CATALOG 7000)

With the new Westinghouse Buying Data, you can select and purchase the proper motor control in half the time.

Data, as presented, is striking in its newness. It's easier to read, easier to understand, and easier to use than any published previously by any manufacturer.

Chances are—if you are a buyer of motors and controls—that you have already received a copy of this new Catalog 7000 by mail. However, if you have not received your copy, write, wire or phone your nearest Westinghouse district office. (Requests will be filled through district offices only — no mailing from Westinghouse headquarters at East Pittsburgh.)

J-60552

Westinghouse
PLANTS IN 25 CITIES . . . OFFICES EVERYWHERE

De-ion Linestarters

Men of Machines

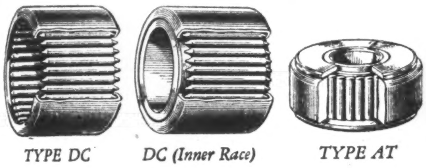
IN HIS NEW position as chief engineer of the Kold-Hold Mfg. Co., Lansing, Mich., H. W. Whitmore will have complete control of design, development and production engineering for the company for the present and postwar eras. Replacing R. H. Swart, Mr. Whitmore holds a bachelor of science degree in electrical engineering from the University of Wisconsin. His experience in the refrigeration industry includes ten years with the General Refrigeration Corp., of Beloit, Wis., and two years with the Automatic Products Co., Milwaukee. He consequently is capably fitted for his new position with the Kold-Hold Co., manufacturer of industrial sub-zero machines.

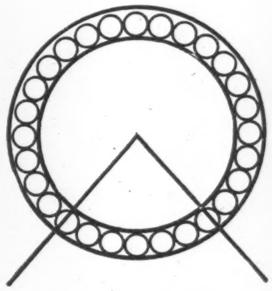
AS CHIEF ENGINEER of The Monarch Machine Tool Co., Sidney, O., Kermit T. Kuck will put to worth-while use his past engineering experience in the machine tool field. Born in New Knoxville, O., Mr. Kuck received his early schooling there, and later was graduated from Ohio State university in 1934 with the degree of Bachelor of Mechanical Engineering. Following graduation he joined The Monarch Machine Tool Co. After an intensive period of training in the shop and the engineering department, Mr. Kuck was moved by the company to Chicago as a project engineer, where he remained until 1940. At this time he was transferred to New Jersey as district manager of the New York-New Jersey territory. His recent appointment as chief engineer of the Monarch organization followed. Mr. Kuck's professional affiliations include the American Society of Tool Engineers and the Engineers' Club of Dayton, O.

AT THE RECENT annual meeting of the American Gear Manufacturers association, Louis R. Botsai was elected president. He is manager of the gearing division of the Nuttall Works, Westinghouse Electric & Mfg. Co. Mr. Botsai's entire business career has been with the Westinghouse company, having become affiliated with the organization upon graduation from Auburn. He served in the company's engineering and sales departments prior to his appointment as manager of the gearing division. In the gear association, Mr. Botsai has been active for a number of years, having served in the capacity of treasurer for four years, and most recently as vice president. He has also served as chairman of numerous committees, including finance and gearmotor, as well as several of the commercial group. His activities in the association as well as in the gearing field will be of considerable assistance to him in the fulfillment of the many and varied duties of his new post. To assist him, Paul W. Christensen has been appointed

(Concluded on Page 180)

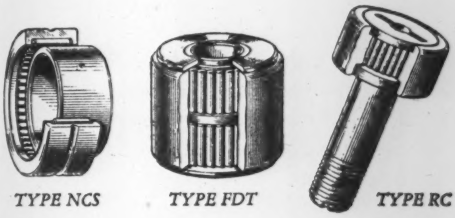
Design of Needle Bearings Eliminates Stress Concentrations





LOAD ZONE

Note how the load is distributed. The large number of the needle rollers which are within the theoretical "load zone" provide many linear inches of contact with bearing surfaces.



FULL COMPLEMENT OF NEEDLE ROLLERS PROVIDES MANY LINEAR INCHES OF CONTACT; DISTRIBUTES LOAD OVER LARGER AREA

Elimination of stress concentrations is one of the outstanding features of all Torrington Needle Bearings which stems from their basic design principle—the use of *a full complement* of small diameter needle rollers.

In operation, this means that the load on the Needle Bearing is widely distributed over a comparatively large number of rollers, and each in turn provides "line" contact with inner and outer race. As illustrated in the accompanying diagram, a large number of needle rollers are always within the "load zone" as the rollers revolve around the shaft. Thus the load is *evenly* distributed and the full capacity* of the bearing may be utilized without fear of concentration of stresses at any given point or on any single component of the bearing. As long as the design load is within the rated capacity of the bearing, the dangers of rapid or uneven wear, of stress fatigue, or of brinelling bearing parts are virtually eliminated.

This feature, coupled with the efficient lubrication of rollers, made possible by the design of the outer raceway which serves to hold the lubricant within the bearing, insures maintenance-free operation and long service life.

For All Design Requirements

From the several types of Torrington Needle Bearings available, the design engineer may select the one best suited to the requirements of a particular application. Load, speed, service conditions, design limitations and cost considerations can be met with full confidence that the right Torrington Needle Bearing, properly installed, will provide long bearing life and freedom from service attention. Let our engineering department work with you in selecting the correct type Needle Bearing for your application. Experience in layout and design of hundreds of different applications will enable our engineers to give practical assistance in meeting your design requirements. More information on the types and sizes of Torrington Needle Bearings together with a more complete description of the advantages of these compact, anti-friction units, will be found in our Catalog No. 30-A available on request. Send for your copy.

**Utilization of full or theoretical capacity of Needle Bearings is dependent upon proper shaft hardness, correct installation and other factors. See Torrington Needle Bearing Catalog for more complete information.*

THE TORRINGTON COMPANY
Established 1866 • Torrington, Conn. • South Bend 21, Ind.
"Makers of Needle Bearings and Needle Bearing Rollers"

New York
Detroit
San Francisco
Toronto
Boston
Cleveland
Chicago
Philadelphia
Seattle
Los Angeles
London, England

TORRINGTON
NEEDLE BEARINGS

(*Concluded from Page* 174)

vice president of the association. Mr. Christensen is president and general manager of the Cincinnati Gear Co., Cincinnati.

◆

Dr. J. T. Rettaliata has been appointed manager of the new research and gas turbine development division of Allis-Chalmers Mfg. Co. Dr. Rettaliata's work has been largely in the development of the gas turbine, which has earned for him annual awards from national engineering and scientific societies for the past three years. As chief engineer in the gas turbine development, J. L. Ray will assist Dr. Rettaliata.

◆

Howard Maxwell, manager, and Milton H. Wells, designing engineer, of the induction motor division, General Electric Co., Schenectady, have retired after 42 and 43 years, respectively, of service. At the same time Charles J. Koch was appointed engineer and Frank D. Phillips, assistant engineer, of the induction motor engineering division.

◆

R. H. Thielemann, who formerly had been associated with the research laboratory of General Electric Co., has been made development engineer of Allegheny Ludlum Steel Corp.

◆

C. F. Thompson has joined Sperry Gyroscope Co., Garden City, L. I., N. Y., as research engineer. He formerly had been assistant chief engineer, Air Associates Inc.

◆

Karl Kramer has been appointed machine designer for the National Advisory Committee for Aeronautics, Cleveland. Formerly he had been product engineer for Leece-Neville Co., Cleveland.

◆

Henry D. Stecher who recently resigned as chief engineer of Weatherhead Co., where he served for about 15 years, has been elected president and general manager of the Romec Pump Co., Elyria, O.

◆

B. C. Erickson, previously connected with Packard Motor Car Co., has become research engineer, Research Laboratories, Borg-Warner Corp., Detroit.

◆

Fred W. Lampe has been made product design and development engineer of CAG Products Inc., Dearborn, Mich. Formerly he had been head of the mechanical engineering section, research laboratory of the Airplane Division, Curtiss-Wright Corp., Buffalo.

◆

Jack S. Beechler is now acting chief engineer with the Firestone Rubber & Metal Products Co., Wyandotte, Mich.

◆

Vernon L. Durrstein has become connected with the Superior Engine division, National Supply Co., Springfield, O., as design engineer. He had been employed as chief engineer at Harnischfeger Corp., Port Washington, Mich.

◆

J. J. Mizer, former chief engineer of Walker Mfg. Co., Racine, Wis., is now a mechanical engineer for Young Radiator Co., Racine, Wis.

◆

Albert L. Stica has been appointed vice president of Multi-Engineering Co. Inc. He is a former member of the engineering staff of Edward G. Budd Mfg. Co.

◆

Jesse G. Vincent, vice president of engineering for Packard Motor Car Co., Detroit, has been awarded a fellowship by the Institute of Aeronautical Sciences at its annual meeting.

◆

Werner E. Lerch has become associated with Ebasco Services Inc., New York city, as diesel engineer in the design and construction division. Mr. Lerch had been a consulting engineer for Douchkess & Lerch.

◆

John W. Horner, formerly test engineer for Guiberson Diesel Engine Co., Dallas, has been made development engineer for Aerojet Engineering Corp., Pasadena, Calif.

◆

Stanley G. Marks has become connected with the engineering staff at Allis-Chalmers Mfg. Co., Springfield Works, Springfield, Ill. He had been associated with Wilcox-Rich division of Eaton Mfg. Co., as assistant engineer.

◆

Edward C. Wells has recently been honored as "Seattle's Young Man of the Year". He is chief engineer of Boeing Aircraft Co. and well known in the field of design and development of four-motored airplanes.

◆

Alva W. Phelps, former assistant vice president of General Motors Corp., has been appointed president of Oliver Farm Equipment Co. He has been in charge of engineering and manufacturing divisions of Frigidaire, Delco Products, and Eastern Aircraft divisions, among others.

◆

L. S. Barksdale, previously chief project engineer at Adel Precision Products Corp., Burbank, Calif., has opened his own engineering office in Glendale, where he will do consulting and development work on mechanical and hydraulic equipment for aircraft.

◆

Harold E. Webb is now an executive assistant in charge of research and development at Kinner Motors Inc., Glendale, Calif. He formerly had been vice president for Adel Precision Products Corp., Burbank, Calif.

When there's "no other way"... look to

AMERICAN SEAMLESS FLEXIBLE TUBING!

THE AIR LINES of this Vaill No. 6 Tube Beading and Flaring Machine are made of American Seamless Flexible Tubing. Four lengths of 3/8" tubing connect two ports on each of two cylinders with the air control valve. A fifth length of 1/2" tubing carries the main air supply to the valve.

Mr. John L. Vaill, president of the Vaill Engineering Company, points out that, while there is no movement in the flexible tubing once it is installed, lack of space made it impossible to make the connections with rigid pipe. The use of preformed copper tube was also considered but discarded as impractical. American Seamless Flexible Tubing, as illustrated, is installed regularly in these units in less than an hour's time.

Whenever you have a problem in conveying liquids or gases, in connecting moving or misaligned parts, isolating vibration or in shielding electric wiring, remember that there is an American Metal Hose product that will solve your problem. Assistance of our Technical Department is available on request. Write for our new 40-page Catalogue, SS-50, describing Seamless Flexible Tubing.

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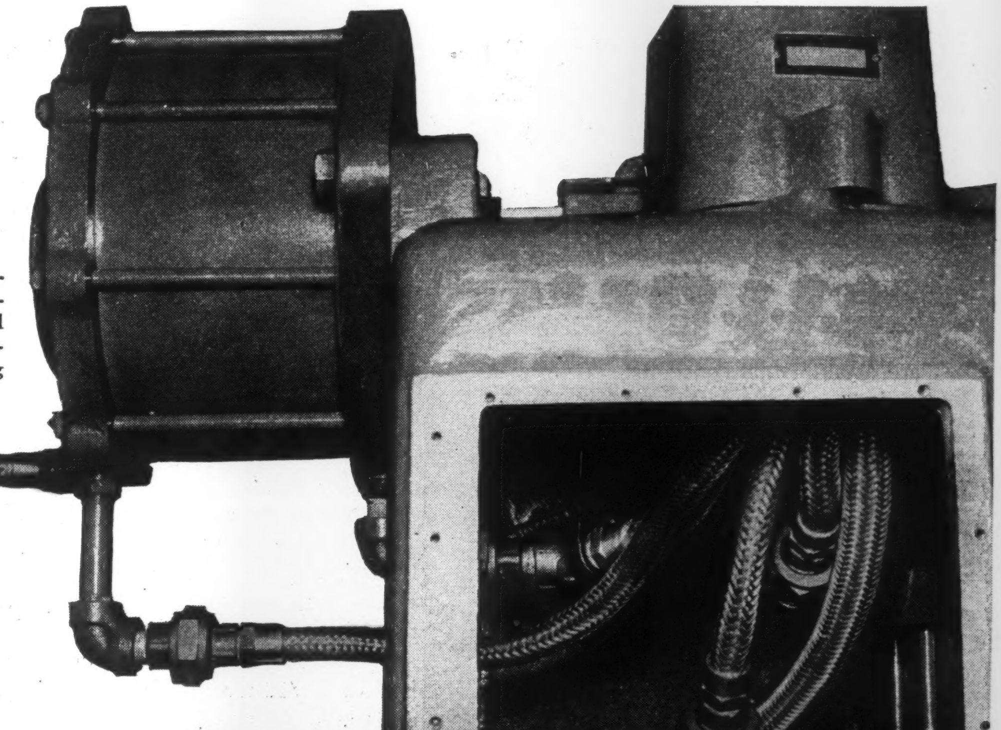


American Seamless Flexible Tubing used in Vaill No. 6 Tube Beading and Flaring Machine.

ANACONDA
from mine to consumer

American Metal Hose

Applying Creep Data

(Concluded from Page 126)

stress is:

$$e=A_1(\epsilon^{S/s_1}-1)[t+T_0(1-\epsilon^{-t/T_1})] \qquad (13)$$

This creep-time-stress relationship involves the determination of four experimental constants in place of two required in the log-log and log methods. For this reason a better agreement with the test data can be expected. A comparison of the various methods for a number of different test results showed that the Soderberg method predicted more accurately the creep-time relationship.[11] It should be noted, however, that there is considerable data to be considered before a definite conclusion can be reached.

Furthermore, it is possible that certain methods will be suitable for particular materials, stress ranges and temperature variations. One of the objections to the Soderberg method is that its increased accuracy is obtained as the result of a more complicated equation. This must sometimes be the case. A creep-stress relation which is not as complicated as Equation 13 but does not assume the linear creep-time variation of the log-log and log methods is the relation proposed by Nadai[10] and discussed in the following:

NADAI'S METHOD: This method assumes a hyperbolic sine relationship between the creep rate and stress and is sometimes called the *hyperbolic sine creep law*. By this relation the creep rate is:

$$C=C_1\,sinh\,S/S_1 \qquad (14)$$

or

$$C=C_1[\epsilon^{S/S_1}-\epsilon^{-S/S_1}] \qquad (15)$$

where C_1 and S_1 are experimental constants.

Compared to the log-log method this method can be extrapolated more accurately to lower stress values. The application of this method has been simplified recently by the use of charts.[12]

Design Stress Variations

The foregoing creep rate-stress or creep-stress relations represent the main ones which have been proposed. Having selected one of these the designer must next determine a design stress by solving the equation selected for the stress S corresponding to an allowable creep or creep rate for the estimated life of the part. This is done for a particular temperature of operation. It is sometimes necessary to consider the influence of temperature change. *Fig.* 6 shows graphs which represent the design stress variation with temperature for various allowable values of the creep. Attempts to express the curves in *Fig.* 6 mathematically have been proposed. One method proposed by Bailey assumes the following relation between the minimum constant creep rate C, the temperature T and the stress S:

$$C=a_1\epsilon^{b_1T}S^n \qquad (16)$$

where a_1, b_1, and n are experimental constants.

Although the log-log method of interpretation is generally used, the problem of extrapolating creep test data is still somewhat controversial. A comparison of the methods of interpretation shows that a considerable difference is possible in the working stress value, depending upon the method used.[13] There is considerable data, however, which supports the log-log method.[5] This article has attempted to review briefly the methods of interpretation for the determination of design stresses. For a further discussion of other factors involved in the interpretation of creep test data the recent papers by McVetty[14] and Gillet[15] are of interest.

EXAMPLE: To explain the application of the foregoing relations the design of a bolt subjected to high temperature accompanied by creep will be considered. The problem is to determine the diameter of the bolt. Axial tensile load remains essentially constant and is equal to 10,000 pounds. Operating temperature is 400 degrees Cent. The steel to be used is a .30 per cent carbon steel with experimental constants $B=4.78\times10^{-37}$ square inches per pound per day, and $n=6.90$ (see Equation 3). Permissible creep rate over a period of 20 years is to be .001-inch per inch per year. Allowable creep rate C in inches per inch per day is $.001/365=2.75\times10^{-6}$. Then by Equation 3 the design stress S is

$$S=(C/B)^{1/n} \text{ or } P/A=(C/B)^{1/n}$$

where P = the total load and A = the cross-sectional area. Then

$$A=P(B/C)^{1/n}=\frac{10{,}000\times(4.78\times10^{-37})^{1/6.9}}{(2.75\times10^{-7})^{1/6.9}}=.49 \text{ sq. in.}$$

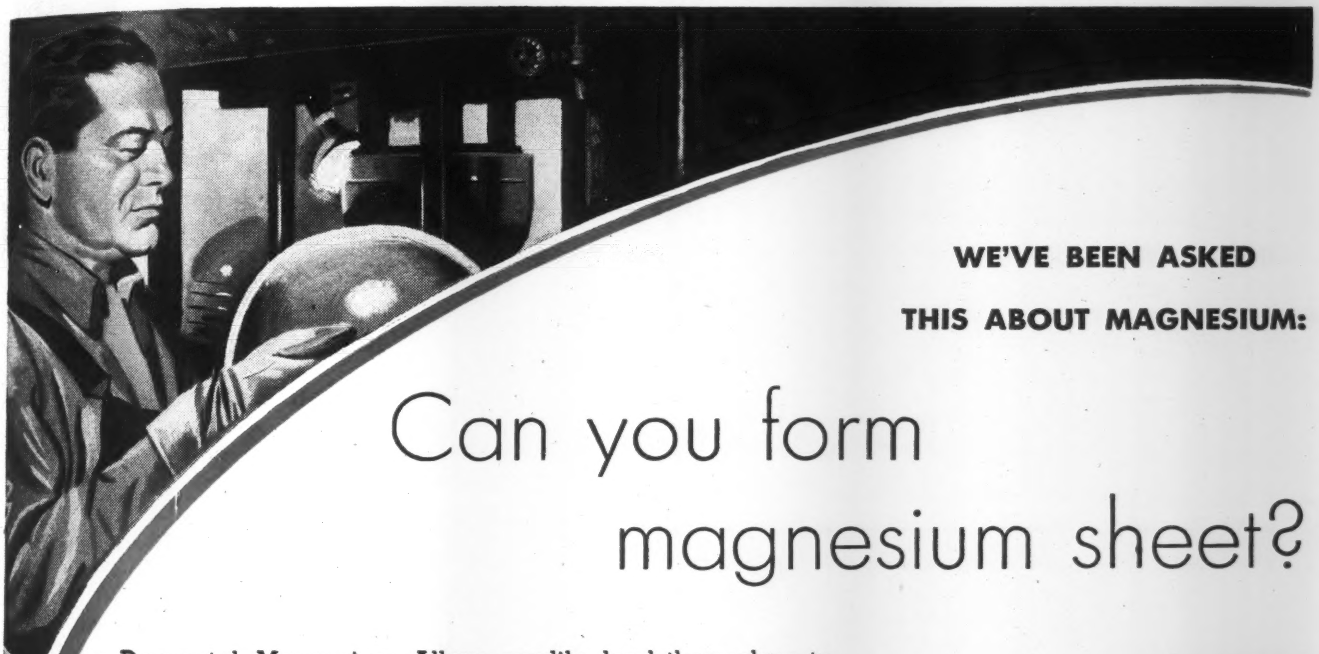
A bolt having this area is about 13/16-inch in diameter. The above calculation does not consider the yield strength. If the yield strength of the steel is 80,000 pounds per square inch and a factor of safety of 2 is used, the area required is $10{,}000/80{,}000=.125$ square inch. For this area the bolt diameter is about 7/16-inch. That is, a bolt of 13/16-inch diameter would be specified for this design.

In the next issue engineering applications involving other kinds of stresses will be discussed.

IN A RECENT SURVEY, over 90 per cent of the people interviewed could not compute correctly the cost of operation per day for a 30-watt lamp when informed of the cost per kilowatt-hour. More significant than this, however, was the fact that each resented having his inability exposed. This brings out the extreme necessity for avoiding technical terms when dealing with the public.

---

11. Report on Comparison of Methods of Interpretation of Creep Test Data prepared for the Turbine Division, Westinghouse Electric and Manufacturing Company, 1937.
12. Nadai, A. and McVetty, P. G.—"Hyperbolic Sine Chart for Estimating Working Stresses of Alloys at Elevated Temperatures", *Proceedings A.S.T.M.*, Vol. 43, 1943, Page 735.
13. Marin, J.—"A Comparison of Methods Used for Interpreting Creep Test Data", *Proceedings A.S.T.M.*, 1937, Page 258.
14. McVetty, P. G.—"Interpretation of Creep Test Data", *Proceedings A.S.T.M.*, Vol. 43, 1943, Page 707.
15. Gillet, H. W.—"Choosing the Right Material", MACHINE DESIGN, Feb., 1944, Page 152.



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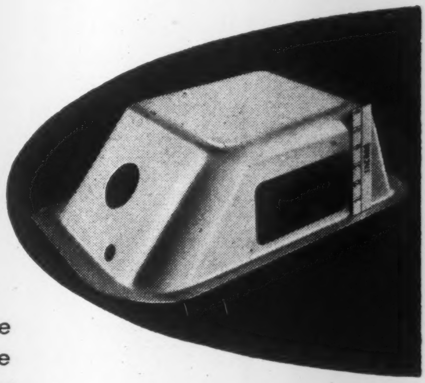
THIS ABOUT MAGNESIUM:

Can you form magnesium sheet?

Dowmetal Magnesium Alloys readily lend themselves to forming, bending and drawing operations. Wherever these fabrication techniques are used, in any field of manufacture where the high strength-weight ratio of the metal is a prime consideration, magnesium is peculiarly valuable.

Not only does it have excellent strength and elastic energy absorption qualities, but it is the lightest of all structural metals.

Dowmetal sheet can ordinarily be formed with the same equipment used for other metals, or with only minor die modifications. Hot forming is generally preferred in working Dowmetal as it permits deep draws in a single operation and eliminates the need of intermediate annealing; too, no allowance for springback is necessary. Cold forming is used for shallow draws and bends of reasonably generous radii.



Dow is currently producing for Bell Aircraft Corporation this weight-saving aircraf pedal-well

Those who have seen the broad scope of the research and quantity-production facilities at the Dow shops and foundry in Bay City, Michigan, know of the progress that has been made in the various methods of Dowmetal fabrication: in sand, permanent mold and die casting; extrusion; forging; sheet, plate and strip; in riveting and welding. In answering inquiries, Dow, as the pioneer and major producer of magnesium, offers the knowledge accumulated during 28 years of intensive research.

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THE METAL OF MOTION

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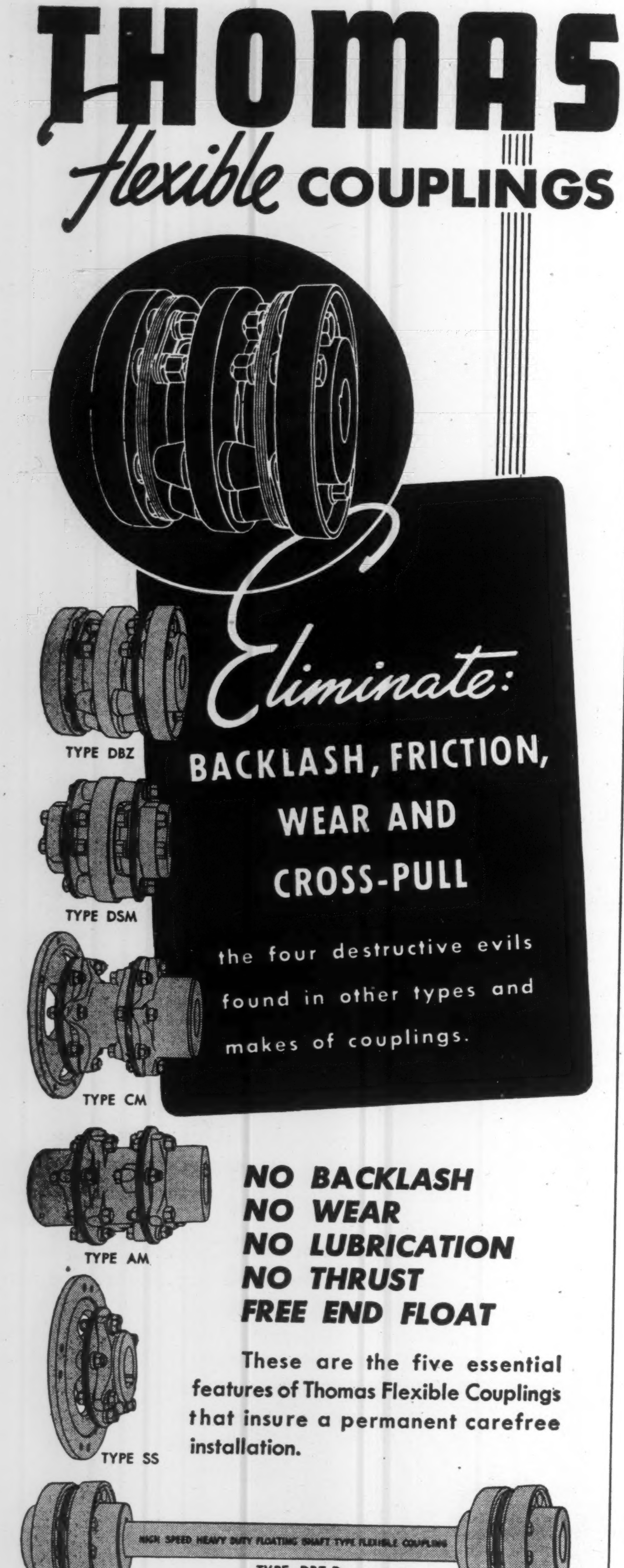

THOMAS *Flexible* COUPLINGS

Eliminate:

BACKLASH, FRICTION, WEAR AND CROSS-PULL

the four destructive evils found in other types and makes of couplings.

TYPE DBZ
TYPE DSM
TYPE CM
TYPE AM
TYPE SS
HIGH SPEED HEAVY DUTY FLOATING SHAFT TYPE FLEXIBLE COUPLINGS
TYPE DBZ-D

NO BACKLASH
NO WEAR
NO LUBRICATION
NO THRUST
FREE END FLOAT

These are the five essential features of Thomas Flexible Couplings that insure a permanent carefree installation.

WRITE FOR COMPLETE ENGINEERING CATALOG

THOMAS FLEXIBLE COUPLING CO.
WARREN, PENNSYLVANIA

BUSINESS AND SALES BRIEFS

SUCCEEDING E. H. Broden is L. F. McGlincy as manager of operations for the Pittsburgh district of American Steel & Wire Co., Cleveland. Mr. Broden had been with the company for forty-eight years, serving as manager of this district since 1933.

◆

Formation of a plastics and chemical sales division to be headed by Herman R. Thies has been announced by Goodyear Tire & Rubber Co.

◆

At the recent annual meeting, Henry W. Jackson was elected president, while Charles D. Addams and John I. Hartman were made vice presidents of Bearings Co. of America, Lancaster, Pa.

◆

Opening of a new Cleveland branch office in the Engineer's building, to serve Kentucky and Ohio with the exception of the Toledo area, has been announced by The Bristol Co., Waterbury 91, Conn. Also announced are the appointments of H. C. Clarke as district manager of the Pittsburgh branch office and G. H. Gaites as regional sales supervisor of Cleveland and Pittsburgh territories, with headquarters at Cleveland.

◆

Brown Instrument Division, Minneapolis-Honeywell Regulator Co., has placed Jack E. MacConville and Al J. McCullough in joint charge of industrial instrument sales at the Cleveland office.

◆

Appointment of N. H. Brodell, previously metallurgical sales engineer, as Cleveland district manager has been announced by Copperweld Steel Co., Warren, O.

◆

With offices at 164 South Central avenue, Los Angeles 12, Arleigh W. Anderson has been named sales representative for the southern California territory of Ross Heater & Mfg. Co. Inc.

◆

Reelection of Norman F. Smith as president of American Brush Manufacturers' Association has been announced. Mr. Smith is vice president and general manager of Osborn Mfg. Co., Cleveland.

◆

According to a recent announcement H. K. Porter Co. has named W. T. Campbell manager of the Philadelphia district office. The position of field service engineer in northern New Jersey for the Chemical Process and Quimby Pump divisions

How Much Cold Can Rubber Stand?

Sub-zero temperatures as low as —120° F. are produced by the machine illustrated. It is a **Deepfreeze** *"Cascade Unit"* manufactured by the Deepfreeze Division of the Motor Products Corporation of North Chicago, Ill.

This modern testing machine is used by The Ohio Rubber Company to predetermine the cold resistance of every rubber and synthetic rubber compound formulated to meet specifications for parts to be subjected to sub-zero temperatures.

Unless properly compounded to resist extreme cold, soft rubber or synthetic rubber can become as hard and brittle as glass, and as easily shattered. Such failures are prevented at "Ohio Rubber" by the use of the machine illustrated.

This is but one of many types of modern testing equipment used by "ORCO" technicians to meet exacting specifications for mechanical molded and extruded rubber parts including special processes for bonding rubber and synthetic rubber to metals and other materials.

Refer *YOUR* specifications to *"ORCO"* for thorough and economical production.

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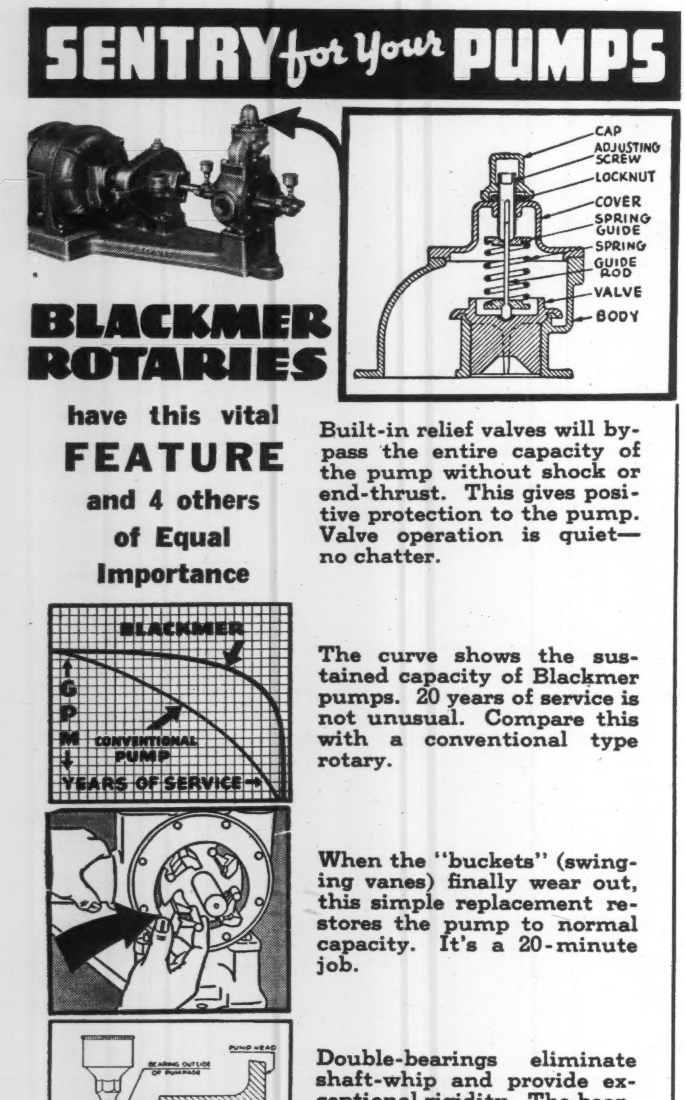

SENTRY for your PUMPS

BLACKMER ROTARIES

have this vital
FEATURE
and 4 others
of Equal
Importance

Built-in relief valves will by-pass the entire capacity of the pump without shock or end-thrust. This gives positive protection to the pump. Valve operation is quiet—no chatter.

The curve shows the sustained capacity of Blackmer pumps. 20 years of service is not unusual. Compare this with a conventional type rotary.

When the "buckets" (swinging vanes) finally wear out, this simple replacement restores the pump to normal capacity. It's a 20-minute job.

Double-bearings eliminate shaft-whip and provide exceptional rigidity. The bearings are located outside the pump casing, away from the pumpage, protected by packing.

For tough jobs, handling corrosive or mildly abrasive liquids, Blackmer pumps are furnished with removable liners. When finally worn out, the liner is replaced and the pump restored to normal capacity. This saves the cost of a new pump.

WE DESIGN & BUILD SPECIAL PUMPS
OUR ENGINEERS ARE AT YOUR SERVICE

POWER PUMPS	HAND PUMPS
5 to 750 GPM.	1½ to 25 GPM.
Pressures to 300 psi.	Pressures to 125 psi. - 56 models.

Write for new Bulletin No. 304—Facts about Rotary Pumps
Blackmer Pump Co., 1977 Century Ave., Grand Rapids 9, Mich.

BLACKMER Rotary PUMPS
"BUCKET DESIGN"—SELF-ADJUSTING FOR WEAR

of the company has been given to J. L. Cunningham. Another new service and engineering office, in the Alliance Bank building, Rochester, N. Y., will be under the direction of recently appointed manager Joseph F. Gaffney. Located in Carew Tower, Cincinnati, a similar office is in charge of R. W. Steves.

◆

Moving of the Cleveland office to 4506 Prospect road has been announced by Allen-Bradley Co., Milwaukee. District manager R. J. Roy will continue to be in charge.

◆

Succeeded by W. F. Carroll as district sales manager for the New York and Pennsylvania districts, H. F. Maxon has been appointed general sales manager, Boston Woven Hose & Rubber Co., Cambridge, Mass. W. I. Lewis has been made assistant general sales manager.

◆

Appointment of Henry N. Wheeler as sales manager for Hartford Electric Steel Corp. and Roxbury Steel Casting Co., Hartford, Conn., has been announced.

◆

Recently the plastics division of Monsanto Chemical Co. has announced the appointment of Carl Whitlock as Detroit branch manager. Previously in the technical service department, Mr. Whitlock will be in charge of plastics materials sales. C. G. Gress, Detroit branch manager since 1940, has been made sales manager of the Resinox department in Springfield, Mass.

◆

Empire Steel Corp., Mansfield, O., has appointed George L. Gaalaas manager of sales of electrical sheets. Clarence F. Beddard has been promoted from assistant sales manager to general sales manager.

◆

Promotion of E. B. Winning from district manager to assistant to the vice president in charge of operations has been announced by Republic Steel Corp., Cleveland. He has been succeeded as district manager by J. L. Hamilton, former assistant district manager.

◆

With headquarters at 420 Board of Trade building, Indianapolis, Stanley G. Disque has been made Indiana sales agent for Follansbee Steel Corp., Pittsburgh. Prior to his appointment, Mr. Disque had been with Republic Steel Corp., Cleveland, in the capacity of district sales manager of the Indianapolis office.

◆

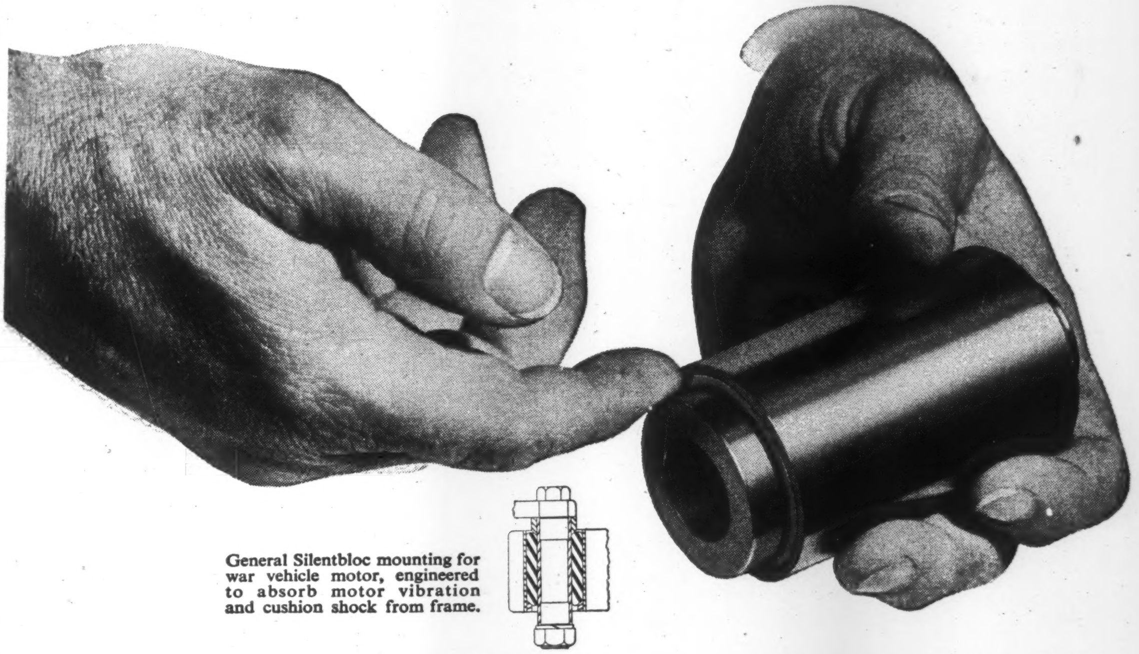
Malkin-Illion Co. has announced that 396 Coit street, Irvington 11, N. J., will be the new location of its offices and factory.

◆

Formerly associated with Lord Mfg. Co. as senior sales engineer, Tom Yates has joined the sales department of Skinner Engine Co., Erie, Pa., where his father had been general manager until his demise.

THE GENERAL TIRE & RUBBER COMPANY OFFERS

Engineered Control of Vibration To Improve Any Moving Product



General Silentbloc mounting for war vehicle motor, engineered to absorb motor vibration and cushion shock from frame.

THE GENERAL SILENTBLOC

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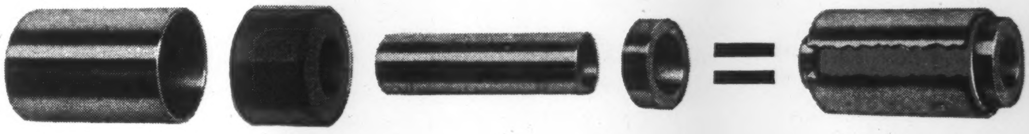

GENERAL SILENTBLOC is made of three main members—1. Straight or flanged tube. 2. Rubber cushion ring. 3. Inner sleeve or solid shaft. Rubber is inserted in tube under high pressure; then sleeve is "shot" through rubber ring with terrific force. Extreme pull of elongated rubber produces indestructible cohesion of parts. Each Silentbloc design is engineered to give needed deflection in all planes.

If your product moves, has working parts or can be harmed by foreign vibration—General Silentbloc can improve its efficiency, lengthen life and lower maintenance.

Silentbloc is not a "piece of rubber" stuck in because it is resilient. These shear-type mountings, bearings and couplings are engineered in shape, size and design to give the exact performance that solves your particular problem.

In its various forms, Silentbloc can do three things:

1. *Damp vibration or insulate against it.*
2. *Give trouble-free torque action.*
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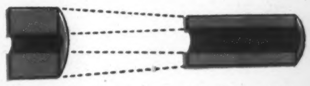
The predictable operation of Silentbloc comes from its patented principle of elongation and confinement of the rubber.



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Our skilled engineers can fit Silentbloc to your job by varying its size and design, the elongation, the distortion of inner and outer diameter of the rubber, the type and modulus of the rubber. The stretched rubber stays alive even when not in motion.

Silentbloc has proved its advantages in many fields—war material, automobiles, aircraft, industrial and home machinery, electrical and electronic equipment. It can improve your product. For fuller information, write The General Tire & Rubber Company, Dept. 70, Wabash, Indiana.

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